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Public Interest Energy Research
CALIFORNIA ENERGY COMMISSION

Building End-Use
Energy Efficiency

EVALUATION OF SMALL COMMERCIAL AIR-CONDITIONING UNITS FOR NORTHERN AND CENTRAL CALIFORNIA

Gray Davis, *Governor*

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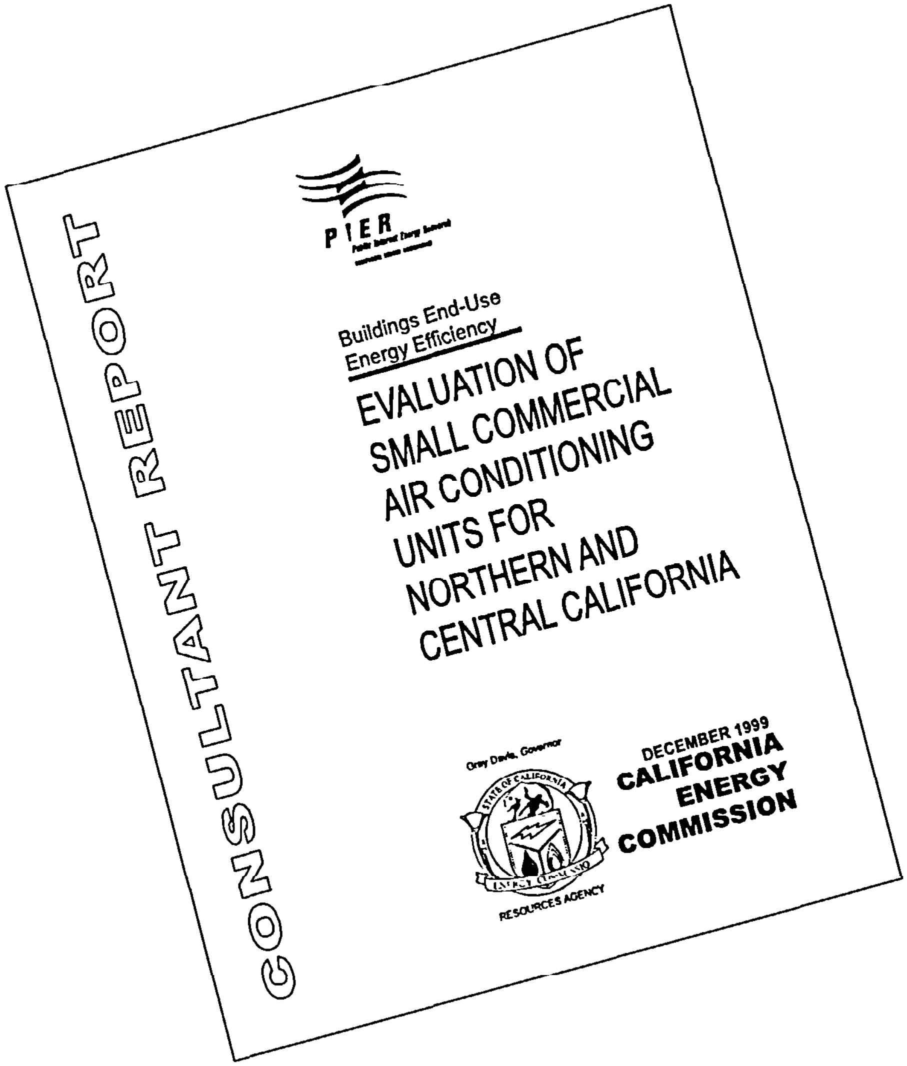

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Legal Notice

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Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to $62 million through the Year 2001 to conduct the most promising public interest energy research by partnering with Research, Development, and Demonstration (RD&D) organizations, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following six RD&D program areas:

- Buildings End-Use Energy Efficiency
- Industrial / Agricultural / Water End-Use Energy Efficiency
- Renewable Energy
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Strategic Energy Research.

In 1998, the Commission awarded approximately $17 million to 39 separate transition RD&D projects covering the five PIER subject areas. These projects were selected to preserve the benefits of the most promising ongoing public interest RD&D efforts conducted by investor-owned utilities prior to the onset of electricity restructuring.

What follows is the final report for the Evaluation of Small Commercial Air Conditioning Units for Northern and Central California project, one of nine projects conducted Pacific Gas and Electric Company. This project contributes to the Buildings End-Use Energy Efficiency program.

For more information on the PIER Program, please visit the Commission's Web site at: http://www.energy.ca.gov/research/index.html or contact the Commission's Publications Unit at 916-654-5200.

Executive Summary

Rooftop packaged air conditioning units are used in many commercial building applications across the United States. Approximately 55 percent of the total annual tonnage of commercial heating, ventilating, and air conditioning (HVAC) equipment sold in the U.S. consists of unitary packaged equipment, the majority of these rooftop units. Current standards require a minimum Energy Efficiency Ratio (EER) of 8.9 Btu / Wh at 95°F outdoor temperature for all units with capacities between 65,000 and 135,000 Btu / h (5.4 to 11.3 tons). The 5 to 10 ton range is the most widespread in California for the small commercial market.

Higher efficiency rooftop air conditioners are on the market. In addition, there is equipment available that can be added to a rooftop unit to improve its efficiency. These higher efficiency options provide a significant potential for energy and demand savings in California. Many parts of California have hot, dry summers that provide opportunity for efficiency improvement not available in more humid climates. Various available technologies can take advantage of these weather conditions to improve efficiency.

To evaluate the potential energy and demand savings of these higher efficiency options, good unbiased performance data over a range of operating conditions are required. While standard Air-Conditioning and Refrigeration Institute (ARI) ratings are available, questions concerning the performance of these systems in off-design conditions exist. Depending on the technology, performance variations at different operating conditions can lead to varying conclusions regarding overall energy and demand requirements in different locations. Performance curves generated through this testing can be used to evaluate the potential economic advantages of various options for rooftop packaged air conditioning systems operating in California's climates.

This project evaluated advanced, small commercial rooftop packaged cooling technologies for operation in California's hot dry climate. It focused on commercially available technologies.

Project Objectives

The specific objectives of this project were to:

- Identify those advanced technologies, which can potentially improve the energy efficiency of air conditioning applications using rooftop packaged air conditioners on small commercial buildings.
- Document the actual performance of some selected advanced technologies through laboratory testing over a range of operating conditions with emphasis on those conditions typical of California's hot dry regions.

Project Outcomes

- Five advanced technology approaches were identified that could potentially result in EERs greater than the minimum national standard of 8.9. They were:
  – Pre-cooling outside air entering the air conditioner
  – Use of an economizer to introduce cooler outside air
  – High efficiency conventional systems
  – Use of an evaporative pre-cooler on the condenser inlet air
  – Refrigerant sub-cooling
- We successfully tested a baseline package unit with an ARI EER rating of 8.9 and two advanced technology units:
  – Baseline unit with evaporative pre-cooler on the condenser
  – High efficiency dual compressor unit.
- Neither the baseline unit nor the high efficiency dual compressor unit could achieve a measured EER that met their published ARI ratings. The baseline EER 8.9 unit had a measured EER of 8.02 at ARI conditions and the high efficiency dual compressor; the EER 11.0 unit had a measured EER of 8.40 at ARI conditions. Much of the shortfall was due to large indoor fan energy use.

Baseline Unit with Evaporative Pre-cooler on the Condenser

- An evaporative pre-cooler used to cool condenser air increased baseline unit EER by seven percent at ARI rating conditions (95°F ambient dry bulb temperature, 40 percent relative humidity). At hot, dry conditions (115°F ambient dry bulb temperature, 15 percent relative humidity), the EER increased by 24 percent as compared with the baseline unit.
- The potential annual energy cost savings of the evaporative pre-cooler on the condenser inlet air technology was estimated to be about 13 percent for one example office building in the Fresno, CA area. However, the installation costs, because of low volume and lack of contractor experience with the technology, results in an extended pay-back period.

High Efficiency Dual Compressor Unit

- The high efficiency dual compressor unit had an EER five percent higher than the baseline unit at ARI rating conditions, but only one percent better than the baseline unit at the hot, dry conditions.
- The Integrated Part Load Value (IPLV) for the high efficiency dual compressor unit was 7.66, significantly below the rated value of 11.6. This indicates reduced performance at part load conditions (single compressor operation). Again this is due to the large percentage of indoor fan energy use.
- Potential energy savings for the high efficiency unit dual compressor unit were more difficult to estimate due to the dual compressor operating mode. Using a simple model with both compressors operating together, the annual energy cost savings was estimated to be about four percent for the same example office building in the Fresno

area. Comparisons using the IPLV and seasonal energy efficiency ratio (SEER) values indicate an annual energy *increase* for the high efficiency dual compressor unit compared to the baseline unit. This is due to the lower system EER during single compressor operation. Depending on the full load and part load performance and the method of compressor control and cycling, this technology may or may not save energy compared to a baseline unit. More work needs to be done to model the performance of this technology in actual installations in order to come up with realistic energy use comparisons.

The table at the end of this section summarizes the test results from this project.

Conclusions

Neither the baseline unit nor the high efficiency unit had measured EERs that met their specified ARI rating. The differences were primarily due to the larger indoor (evaporator) fan power use on units we tested as compared to the rated units. Because this fan uses a relatively large percentage of the total power, differences in its power use due to different static pressures, fan speeds, or different model motors or drives can significantly effect overall unit EER. When comparing the performance of these units based on their ARI ratings, it is important to consider the actual expected installation conditions for external static pressure and flow requirements, and consider the actual fan required to meet those conditions. The actual EER may be considerably lower than the ARI rating when this is done.

The addition of an evaporative pre-cooler on the condenser resulted in improvements in energy efficiency and demand, and increases in capacity over a wide range of climate conditions (temperature and humidity). Compared to the baseline unit at the standard rating conditions, the capacity increased by four percent, the electric input decreased by four percent, and the resulting EER increased by seven percent. For the maximum operating condition test, which has a much lower relative humidity, the capacity increased by 15 percent, the electric input decreased by seven percent, and the resulting EER increased by 24 percent. However, under some conditions of high humidity the performance actually decreased, which makes its application less attractive for some higher humidity regions, such as along the coast.

The use of a high efficiency dual compressor unit also resulted in improvements in energy efficiency and demand as compared to the baseline, standard efficiency unit. At the standard rating conditions, the EER of the high efficiency dual compressor unit was five percent higher and electric demand was 11 percent lower. But cooling capacity of the high efficiency dual compressor unit was lower by seven percent. At the hot, dry maximum operating condition, the high efficiency dual compressor unit showed relatively small improvement over the baseline unit. The EER was higher by one percent, electric demand, lower by two percent. Again, cooling capacity was lower, this time by about one percent.

Estimated energy and demand savings and resulting payback periods from these higher efficiency options depend largely on building type and location and the resulting cooling load. For a specific example office building in Fresno, CA, the baseline unit with evaporative pre-cooler on the condenser was estimated to save about 13 percent of the cooling energy used compared to the baseline unit.

Energy savings for the high efficiency dual compressor unit were more difficult to estimate because of the dual compressor operation.

The estimated extra installed cost for the baseline unit with evaporative pre-cooler on the condenser was about $2,700, compared to about $1,000 extra for the high efficiency dual compressor model. The total energy and demand savings for the baseline unit with evaporative pre-cooler on the condenser was at least twice as great, so the simple pay-back periods were not significantly different (assuming four percent energy savings for the high efficiency unit). Some example pay-back calculations for Fresno, CA gave values ranging from 6 to 13 years, depending on the cooling load. It may make the most sense to install an evaporative pre-cooler on the condenser as an upgrade to existing systems, while purchasing high efficiency dual compressor units for new installations. By bringing the installed costs of the evaporative pre-cooler on the condenser unit down, or increasing the performance of the high efficiency units, the pay-back periods for these technologies would become more acceptable.

Benefits to California

The use of an evaporative pre-cooler on the condenser inlet air, and use of high efficiency dual compressor technologies provide both energy and demand savings for California's small commercial customers. If an average of five percent improvement in efficiency was achieved across all installations, then there is the potential for 170 GWh / yr of savings in California (based on an estimated 3400 GWh / yr of electric energy used by packaged air conditioners on small commercial buildings). The potential demand reduction can be on the order of 100 MW if even the minimum two percent demand reduction under very hot conditions is applied to the estimated three million tons of installed packaged unit cooling capacity on small commercial buildings in California.

While these savings are potentially significant, the savings achievable through the market adoption of more advanced technologies and better design practices are much larger. Improved design practices can decrease fan power by half and thereby reduce individual rooftop package unit demand by 1 kW. Because fans operate year around, the kWh savings achieved by a 1 kW reduction in fan power will be larger than the five percent savings from the two technologies evaluated in this project. The benefit of this research to California will be captured when equipment manufacturers produce equipment specifically designed to operate efficiently in hot dry climates and engineers design low static air distribution systems.

Recommendations

Although some estimates were made, more work needs to be done to better compare the annual energy use of various higher efficiency options based on standard laboratory tests, either through computer models or improved annualized performance factors analogous to the SEER rating.

Additional work needs to be done to address the issue of indoor fan energy use. There needs to be better clarification on how overall performance rating values are affected by the actual fans incorporated into a particular unit. This information should be published in a manner clear to users comparing performance ratings. Efforts should be made to reduce indoor fan energy use by incorporating higher efficiency or variable speed fans, or other alternatives.

Additional performance testing of other high efficiency options for rooftop packaged units should be done and compared to those already evaluated.

Summary of All Test Results

ARI Test Designation	Parameter	Test Unit #1 Baseline Unit	Test Unit #2 Baseline Unit with Evaporative Condenser Pre-cooler [5]	Test Unit #3 High Efficiency Dual Compressor Unit
A	Capacity (Btu/hr)	88,800	92,200	82,900
	(Tons)	7.40	7.69	6.91
Inside: 80°F DB, 67° WB	Power (kW)	11.1	10.7	9.9
Outside: 95°F DB, 75°F WB[1]	EER (Btu/Wh)	8.02	8.58	8.40
B	Capacity (Btu/hr)	96,300	99,600	84,500
	(Tons)	8.03	8.30	7.05
Inside: 80°F DB, 67° WB	Power (kW)	10.4	10.1	8.9
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	9.28	9.90	9.54
C	Capacity (Btu/hr)	87,900	91,700	78,900
	(Tons)	7.32	7.64	6.58
Inside: 80°F DB, 57°F WB[2]	Power (kW)	10.2	9.9	8.9
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	8.61	9.28	8.89
D[3]	Capacity (Btu/hr)	16,500	17,500	14,500
	(Ton-hr/hr)	1.37	1.46	1.21
Inside: 80°F DB, 57°F WB[2]	Power (kWh/hr)	2.2	2.1	1.8
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	7.49	8.30	7.88
Maximum Operating Conditions	Capacity (Btu/hr)	76,000	87,600	74,900
	(Tons)	6.33	7.30	6.24
Inside: 80°F DB, 67° WB	Power (kW)	12.1	11.3	11.8
Outside: 115°F DB, 75°F WB[1]	EER (Btu/Wh)	6.26	7.78	6.34
Low Temperature Operation	Capacity (Btu/hr)	86,700		73,500
	(Tons)	7.23		6.13
Inside: 67°F DB, 57°F WB	Power (kW)	9.1		8.0
Outside: 67°F DB, 57°F WB[1]	EER (Btu/Wh)	9.50		9.21
Test Averages	SEER[4] (Btu/Wh)	8.54	9.26	7.30 (bin)
	IPLV (Btu/Wh)			7.66
	Evaporator Airflow (cfm)	3,380	3,320	2,910
	Condenser Airflow (cfm)	5,210	4,770	6,720
	Evaporator Fan Power (kW)	1.83	1.78	2.30
	Condenser Fan Power (kW)	0.64	0.67	0.62
	Pre-Cooler Effectiveness		51%	
Manufacturer's Specifications	Capacity (Btu/hr)	86,000	93,600	90,000
	(Tons)	7.2	7.8	7.5
(At ARI Rating Conditions)	Power (kW)	9.7	9.5	8.2
	EER (Btu/Wh)	8.9	9.9	11.0
	IPLV (Btu/Wh)			11.6
	Pre-Cooler Effectiveness		60%	

[1] Outside wet bulb condition was not maintained for test units #1 and #3 since they do not involve evaporation.
[2] The low inside wet bulb condition could not always be achieved; coil was likely wet.
[3] Test "D" is a cyclic test in which the unit is on for 6 minutes and off for 24 minutes. For the dual compressor unit, the results given are for both compressors cycling together. Single compressor results are given in the main body of the report.
[4] Seasonal Energy Efficiency Ratio (SEER) calculated for comparison purposes. It is normally only used as a rating number for units having capacities less than 65,000 Btu / hr.
[5] The manufacturer's performance specifications for this system were estimated by combining the evaporative pre-cooler effectiveness specifications and the baseline unit specifications.

Abstract

The objective of this project was to identify and evaluate technologies that could improve the energy efficiency of air conditioning applications using rooftop packaged units on small commercial buildings. The actual performance of selected technologies was documented through laboratory testing. Based on an evaluation of typical sizes and ranges of performance currently available, three technologies were selected for evaluation. The first was a standard efficiency unit meeting the minimum performance requirements used as a baseline unit for comparison to the other technologies. The second was the baseline unit with the addition of an evaporative pre-cooler on the condenser. The third was a high efficiency dual compressor unit. All had 7½ ton nominal capacities.

To test the units, an existing Pacific Gas & Electric test facility was modified. Test instrumentation was installed and procedures developed to test in close agreement with ASHRAE standards. The three technologies were then tested over a wide range of operating conditions to fully characterize their performance.

Performance results are presented for the units, giving capacity, electrical power demand, and energy efficiency ratio (EER) over the range of operating conditions. The following table summarizes the performance comparisons at two specific conditions. These two conditions represent the standard ARI rating conditions, and hot, dry extremes found in California's central valley areas. More detailed information may be found in Table 12. Values are rounded to the nearest percent.

Technology Compared to Baseline:	Nominal ARI Rating Conditions (95/75°F outdoor dry bulb/wet bulb temperatures)	Hot, Dry Conditions (115/75°F outdoor dry bulb/wet bulb temperatures)
Evaporative Pre-cooler on the Condenser		
Increase in EER	7%	24%
Decrease in Electric Demand	4%	7%
Change in Cooling Capacity	+4%	+15%
High Efficiency Dual Compressor		
Increase in EER	5%	1%
Decrease in Electric Demand	11%	2%
Change in Cooling Capacity	-7%	-1%

1.0 Introduction

1.1 Background

Rooftop packaged air conditioning units are used in many commercial building applications across the United States. Approximately 55 percent of the total annual tonnage of commercial heating, ventilation and air condition (HVAC) equipment sold in the U.S. consists of unitary packaged equipment, the majority of these rooftop units. Current standards require a minimum Energy Efficiency Ratio (EER) of 8.9 Btu / Wh at 95°F outdoor temperature for all units with capacities between 65,000 and 135,000 Btu / h (5.4 to 11.3 tons). The 5 to 10 ton range is the most widespread in California for the small commercial market.

Higher efficiency rooftop air conditioners are on the market. In addition, there is equipment available that can be added to a rooftop unit to improve its efficiency. These higher efficiency options provide a significant potential for energy and demand savings in California. Many parts of California have hot, dry summers that provide opportunity for efficiency improvement not available in more humid climates. Various available technologies can take advantage of these weather conditions to improve efficiency.

To evaluate the potential energy and demand savings of these higher efficiency options, good, unbiased performance data over a range of operating conditions are required. While standard Air-Conditioning and Refrigeration Institute (ARI) ratings are available, questions concerning the performance of these systems at off-design conditions exist. Depending on the technology, performance variations at different operating conditions can lead to varying conclusions regarding overall energy and demand requirements in different locations. Performance curves generated through this testing can be used to evaluate the potential economic advantages of various existing options for rooftop packaged air conditioning systems operating in California’s climates.

This project evaluated advanced, small commercial rooftop packaged cooling technologies for operation in California’s hot dry climate. It focused on commercially available technologies.

1.2 Project Objectives

The objectives of this project were to:

- Identify those advanced technologies, which can potentially improve the energy efficiency of air conditioning applications using rooftop packaged air conditioners on small commercial buildings.
- Document the actual performance of some selected advanced technologies through laboratory testing over a range of operating conditions with emphasis on those conditions typical of California’s hot dry regions.

1.3 Project Approach

To accomplish these objectives, the project was divided into the following tasks. Tasks 1 and 2 were administrative tasks.

Task 3 – Research Available Technologies

Pacific Gas & Electric (PG&E) assessed the available technologies for small commercial rooftop packaged air conditioning systems. Specifically, this involved literature searches, vendor discussions, and discussions with other HVAC experts and included an assessment of typical sizes of units, as well as the range of performance available. Based on this investigation, one to three technologies would be selected for further testing and evaluation. We selected three technologies to evaluate further through laboratory testing.

Task 4 – Develop Test and Evaluation Plan

We developed a test and evaluation plan for the technologies selected in Task 3. The plan included determining test facility and instrumentation requirements, and developing test conditions specification and detailed procedures for testing under a range of operating conditions. The existing PG&E test facility had been developed for residential-sized units (3 ton). Detailed evaluation and re-design were required to adapt the facility for evaluations of larger units. This task resulted in a facility design and test plan to allow testing of units up to 10 tons in capacity, over a wide range of operating conditions.

Task 5 – Modify Test Facility as per Plan

This Task modified the test facility based on the specification developed in Task 4. This included modification of the existing test buildings, and procurement of additional instrumentation and equipment as needed. After the initial design requirements were specified, a final design of the test facility modifications was performed. A contractor was hired for the final design and completion of the modification. As much of the existing facility and equipment as possible were re-used. Concurrently, the data acquisition system was upgraded to provide the measurements required by the test plan. After completion of the facility modification, the instrumentation was installed and calibrated. A start-up period for debugging and final test preparations followed.

Task 6 & 7 – Procure, Install, and Test Selected Technologies

During Task 6 we procured the small commercial units specified in Task 3. Task 7 involved the installation, start-up, and testing of each selected technology. This included installation and start-up of the unit under test, as well as associated instrumentation for that test. There are slight differences between technologies concerning the exact measurement parameters, so we modified the test plan as necessary for each technology.

The test units were installed and tested one at a time. Tests were performed over a range of operating conditions by varying the condenser inlet air temperature (outdoor air temperature), the evaporator inlet air temperatures (return or indoor air temperature), and the evaporator air flow rate. Most tests were performed under steady-state conditions, but some cycling tests were also performed. Below is a summary of the range of nominal test conditions attempted:

- Outdoor Side Condenser Entering Air Dry bulb Temperature: 55°F - 115°F
- Outdoor Side Condenser Entering Air Wet bulb Temperature: 60°F - 95°F *
- Indoor Side Evaporator Entering Air Dry bulb Temperature: 55°F - 80°F
- Indoor Side Evaporator Entering Air Wet bulb Temperature: 46°F - 75°F
- Evaporator Air Flow Rate controlled by varying external static pressure

* Note: outdoor side wet bulb temperature was only controlled during tests on the unit with an evaporative pre-cooler on the condenser.

A complete matrix of test points is included in the Test and Evaluation Plan (Appendix II). All test conditions listed in the plan could not be achieved for all test units due to test facility limitations described later in the report. Tests under nominal ARI rating conditions were included in the test matrix.

Task 8: Perform Data Analysis

We performed the final data reduction and analysis after all testing and final instrument calibrations were completed. The test results were tabulated and plotted in terms of capacity (tons), power (kW), and EER (Btu / Wh) as a function of various operating conditions.

Task 9: Develop Technology Transfer Plan

We developed an appropriate plan to use these test results in market transformation programs, including evaluation of any follow-on work that might be needed to do so.

Task 10: Final Report and Meeting (combined previous Tasks 10 and 11)

We produced the final documentation of this project in accordance with California Energy Commission requirements and met with the Commission as necessary to present and discuss the results.

1.4 Evaluation of Technologies

Various alternative systems for packaged rooftop air conditions were evaluated for the project. Appendix I provides detailed discussion of the evaluation as well as the Interim Report from Task 3. This Section provides a brief overview of the equipment.

The investigation highlighted the following technologies for consideration:

- Pre-cooling outside air entering the air conditioner using an indirect or combination of indirect and direct evaporative cooling equipment, heat wheel, or heat pipes.
- Use of an economizer to introduce cooler outside air when available in order to reduce cooling load.
- High efficiency systems, which may use combinations of higher efficiency compressors, multiple smaller compressors, and increased evaporator and condenser surface areas.
- Use of an evaporative pre-cooler on the condenser to lower the effective outdoor air temperature entering the condenser.
- Use of refrigerant sub-cooling to increase evaporator capacity while potentially decreasing compressor electrical power.

1.5 Technologies Selected

Based on a number of considerations, three technologies were selected for further evaluation for this project. These were:

- Baseline unit (Test Unit #1): a standard 7½ ton unit just meeting the minimum federally mandated performance requirements
- Baseline unit with evaporator pre-cooler on the condenser (Test Unit #2): a standard 7½ ton unit with the addition of an evaporative pre-cooler on the condenser
- High efficiency dual compressor unit (Test Unit #3): a higher efficiency dual compressor 7½ ton unit

Both the selected standard unit and the high efficiency dual compressor unit were rated at a nominal capacity of 7½ tons. Appendix V gives more detailed specifications on the units.

Table 1 summarizes the specifications for the units.

Table 1. Summary of Test Unit Performance Specifications

	Test Unit #1 Baseline Unit	Test Unit #2 Baseline Unit with Evaporative Pre-cooler on the Condensor[2]	Test Unit #3 High Efficiency Dual Compressor Unit
Capacity (Btu/hr)	86,000	93,600	90,000
(Tons)	7.2	7.8	7.5
Power (kW)	9.7	9.5	8.2
EER (Btu/Wh)	8.9	9.9	11.0
IPLV (Btu/Wh)			11.6

Notes:

[1] These are "high-heat" models which include a gas furnace, although these were unused during the tests.

[2] Test Unit #2 performance values were estimated using an effectiveness specification of 60 percent, and calculating the condenser inlet air temperature at the outdoor conditions for the ARI "A" test condition (reduced condenser inlet air temp = 83°F). The manufacturer's performance specifications for the baseline unit were then used to estimate the system performance at this reduced condenser inlet air temperature. Additional power due to the water pump and added condenser resistance was included.

2.0 Discussion

2.1 Research into Available Technologies

The first task of this project identified and assessed technologies with the potential to improve the energy efficiency of packaged rooftop air conditioners in small commercial buildings. The assessment included the energy efficiency, cost, and availability of these technologies. There was also an assessment of the most appropriate size unit for further evaluation. Relevant literature was reviewed, and discussions were held with vendors and industry experts.

The final selection of an appropriate size unit to evaluate was not clear-cut. We focused on units in the 5 to 10 ton range, but available shipment data were not easily separated into individual unit sizes, residential versus small commercial application, or distribution by state. Opinions gathered from a number of experts also varied. Based on this available information, arguments could be made for the 5 ton, 7½ ton, or even 10 ton or larger size as the most appropriate for evaluation.

We chose to evaluate the 7½ ton size for several reasons. We felt that this size unit allowed us to evaluate the types of technologies most appropriate for the current small commercial applications in California. It is the smallest capacity unit that can be purchased with multiple compressors at this time, is available in higher efficiency models, and can utilize options such as condenser pre-cooling and economizers.

As a basis for comparison, we chose to first evaluate a baseline unit that met the minimum energy efficiency requirements for the 7½ ton size (EER of 8.9 at rating conditions). As a relatively low-cost improvement, we next chose to evaluate the performance of this same unit with the addition of an evaporative pre-cooler on the condenser. This was specified to increase performance 10 percent or more, depending on outside conditions. Finally, we selected a dual compressor unit one of the efficiency ratings in this size (EER of 11.0 at rated conditions). All of the units are commercially available from major suppliers.

Appendix I provides a more detailed discussion of the assessment and selection of these technologies.

2.2 Test and Evaluation Plan

The performance of these units is a function of several operating parameters, and it is not possible to give one performance result without specifying the specific operating conditions. ARI Standard 210/240-94 specifies a set of specific conditions for standard rating purposes. In addition, tests over a wider range of conditions allow creation of a set of performance curves for a unit, or generically for a technology. These curves could then be incorporated into energy simulation models to predict and compare the performance of these systems in actual buildings in various locations.

Appendix II gives details about the test plan for this project, including the range of operating conditions that were attempted. Table 2 summarizes these operating conditions:

Table 2. Operating Conditions for Standard Rating and Performance Tests (ARI)

		INDOOR UNIT Air Entering			OUTDOOR UNIT Air Entering		
	TEST	DB °F	WB °F	(RH) (%)	DB °F	WB[1] °F	(RH) (%)
COOLING	Standard Rating Conditions "A" Cooling, Steady State	80	67	(51%)	95	75	(40%)
	"B" Cooling, Steady State	80	67	(51%)	82	65	(40%)
	"C" Cooling, Steady State, Dry Coil	80	57	(22%)	82	65	(40%)
	"D" Cooling Cyclic, Dry Coil	80	57	(22%)	82	65	(40%)
	Maximum Operating Conditions	80	67	(51%)	115	75	(15%)
	Low Temperature Operation	67	57	(54%)	67	57	(54%)
	Part Load Conditions (IPLV)	80	67	(51%)	80	67	(51%)

[1] The wet bulb temperature condition is required only when testing air cooled condensers which evaporate condensate, or when testing with condensers using evaporation for supplemental cooling

To develop performance curves for these technologies, an additional range of operating conditions was specified. These are summarized below:

Table 3. Operating Conditions for Sensitivity Testing

		INDOOR UNIT Air Entering			OUTDOOR UNIT Air Entering					
	TEST	DB °F	WB° F	(RH) (%)	DB °F					
COOLING	Steady State	80	75	(80%)	115	105	95	82	67	55
			67	(51%)	115[M]	105	95[A]	82[B]	67	55
			57	(22%)	115	105	95	82[C]	67	55
		67	57	(54%)					67[L]	55
		55	46	(50%)						55

Superscripts indicate ARI Standard Test Conditions (Tests A, B, C, Maximum and Low Temperature).

In addition to the steady-state tests listed above, we were interested in evaluating the effects of cycling on performance. Two variables need to be specified for a cycle test, the total period of time for one on/off cycle, and the percentage of time during that cycle that the unit is on (or off). The following table summarizes the planned cycling tests:

Table 4. Cycling Test Plan

Example Limits
Minimum Off Time: 3 minutes
Minimum On Time: 5 minutes

Part Load Ratio	Time On per Cycle	Time Off per Cycle	Period (Min./Cycle)	Frequency (Cycles/Hour)
100%[1]	60 min.	0 min.	60	1
75%	9 min.	3 min.	12	5
50%	7½ min.	7½ min.	15	4
25%	5 min.	15 min.	20	3
20%[2]	6 min.	24 min.	30	2

[1] ARI Standard Test "C" [2] ARI Standard Cycling Test "D"

2.3 Test Facility Modifications

In order to evaluate the selected technologies, modifications to an existing PG&E test facility were required. The facility originally had one test room, adjacent to another building that had been used for other work in the past but which was presently unused. In order to test the technologies according to the plan described in Appendix II, two rooms were required, with additional conditioning capabilities. The unused room (labeled "Room A") was designated as the "outdoor room", and the previous test room (labeled "Room B") was designated as the "indoor room".

Room A needed several structural modifications, including removal of a support beam inside the room, and raising the ceiling height several feet. The foundation was strengthened to handle anticipated weights of test equipment. Large south-facing windows (previously used for solar energy tests) were removed and replaced with a roll-up door for installing and removing test equipment. The conditioning equipment, ducting, and measurement and control equipment were added.

Room B did not require significant modifications. The conditioning equipment (blower, heater, humidifier) located outside of the building were re-used, but additional ducting for removing and supplying conditioned air to the room was added. No provision was added at this time for cooling or dehumidifying the supply air, since the discharge of the test unit could be recirculated and reheated or humidified for many of the test points.

Both test rooms included airflow measurement stations, including variable speed booster fans to compensate for the added resistance. Also, supply and return ducts to the test unit and instrument wire conduits were connected between the two rooms.

Figure 1 is a diagram of the modified test facility:

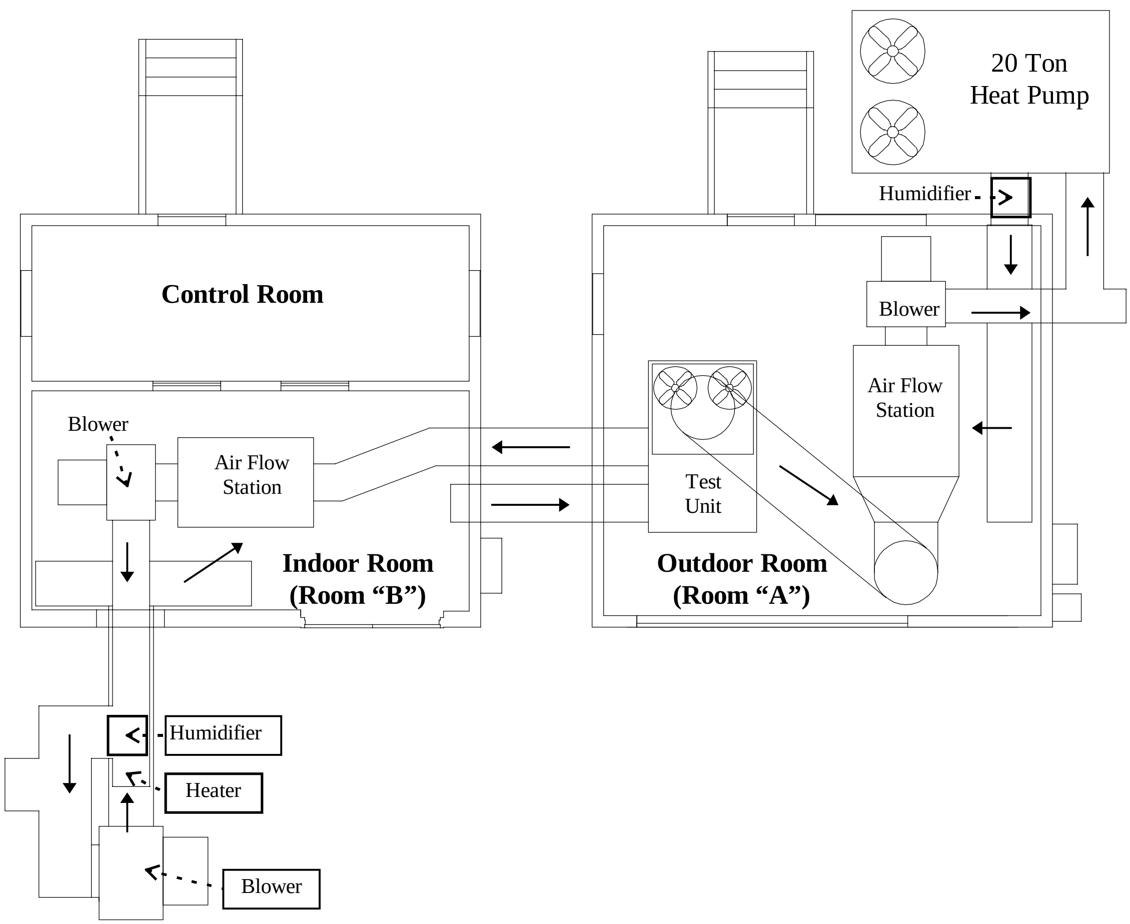

Figure 1. Schematic of PG&E Test Facility

A summary of the test facility specifications follows:

Outdoor Room (Room A):

Approximately 20 ft x 18 ft (360 square feet)

20 ton heat pump conditioning unit with 72 kW electric heater

80 lb / hr electric steam humidifier

Air flow nozzle station (11,000 cfm max.)

7½ hp variable speed blower (11,000 cfm @ 2.25” static pressure)

Indoor Room (Room B):

Approximately 20 ft x 10 ft (200 square feet)

100 kW electric heating unit

150 lb / hr electric steam humidifier

3 hp blower (6500 cfm @ 2” static pressure)

Air flow nozzle station (5,000 cfm max.)

5 hp variable speed blower (5000 cfm @ 3” static pressure)

Some compromises had to be made in the modification of the test facility to meet the constraints of time and budget. Thus, there were some design features that were left out, and some test conditions could not be met under certain ambient conditions. Over the course of testing, a wish list of added features to improve the operation of the facility was developed, and some of its highlights are as follows:

- Indoor Room Cooling / Dehumidifying. Probably the biggest problem in the operation of the facility was trying to achieve low enough humidity to conduct dry coil tests. Although the climate in the area of the test facility is fairly dry, it was often not dry enough. What is needed is the addition of a cooling / dehumidifying coil to the room conditioning apparatus. This is particularly needed for the dry coil cycling tests that must be conducted with 100 percent outside air, and must be dryer and cooler than the test conditions. The only modification done to the indoor room conditioning system as part of this project was the addition of duct work and dampers to allow for recirculation.
- Steam Humidifiers. To increase the humidity of the rooms, steam at atmospheric pressure is added to the room conditioning supply streams. The small steam humidifiers that are used in this facility (both the original one on the indoor room and the new one on the outdoor room) work by passing an electric current between submerged electrodes in a container of tap water. Periodically, a valve is opened to discharge built up salts and other solids, and the lost water is replaced by an additional inflow of cold, fresh water. This not only decreases the temperature of the water in the container; it also lowers its conductivity. The result is that the output of these humidifiers is not constant, resulting in an oscillating humidity in the conditioned

space. It would be better to have a small boiler for producing low pressure steam that can be injected into the room conditioning systems at a controlled rate.

- Outdoor Room Access Door. The access door that was installed in the modified outdoor room is a roll-up door. The original designs for the room called for removable door panels, which would only need to be opened when a test unit was being moved in or out. The roll up door was installed as a compromise for simplicity and time. However, it created two problems. The first was that it was not air-tight or well insulated. The fix for this was to cover the outside of the door with insulating panels during testing, basically recreating the effect of having removable panels. The second problem was that the roll up reel extended out from the wall into the room, which interfered with the positioning of the condenser outlet air duct (see next bullet).
- Condenser Outlet Air Duct. The original design for the condenser outlet duct was a straight section up to the ceiling, across the room, and down to the air flow chamber inlet. The additional height gained by raising the ceiling in the room allows for a greater length of vertical duct out of the test unit, and creates space for test units with bottom supply and return ducts. The outlet temperature measurement could also be placed in this vertical duct, rather than its current location in the cross-over. Also, flexible sections were to be used to more easily allow testing of units of different heights. However, the addition of the roll-up door reel interfered with this design. Review of the ducting arrangement should be done to determine if changes could be made to approach the original design intent.
- Mixing Devices. Although included in the original planning drawings, devices for thoroughly mixing the air leaving the test unit (both on the evaporator and condenser side) were not created and installed. While using a grid of several temperature measurements and averaging takes care of much of the problem of having different temperature regions, a mixing device will ensure consistency for both the temperature and sampling humidity measurements. Mixing devices should also be installed in the room conditioning air supply ducts to allow for better control of the supply air.
- Power Measurement. Some problems were found with some of the watt transducers and current transformers (CTs) located in the test unit to measure individual component power use. There appeared to be a temperature sensitivity that was not anticipated. These CTs and watt transducers should be moved to less extreme temperature locations, or devices less sensitive to these conditions should be obtained.

Future modifications to the test facility or to the measurement system may be made incrementally as permitted by time and budget and as required by future testing applications.

2.4 Measurement System

The instrumentation and data acquisition system for the test facility was developed based on the needs for high accuracy, repeatability, reasonable scan rate, and high channel capacity (which allowed for redundant measurements in key locations). The data acquisition system would also be required to provide feedback control signals to the room conditioning systems and control of the test unit, which required digital and analog output capability. The resulting data acquisition system actually consists of a number of sub-systems and individual instruments that tie into a central personal computer.

Air flow rates were measured using air flow stations constructed according to the designs outlined in ASHRAE and ARI standards. The flow stations consist of a large chamber with a central partition containing a number of air flow nozzles. The size of the chamber is large so that the velocity of the air upstream of the nozzles is negligible in comparison with the velocity through the nozzles. The boxes were equipped with baffle plates to condition the entering and leaving air streams. Individual nozzles can be capped as needed to produce an adequate differential pressure at the operating flow rate (specifically between 0.5 and 3.0 inches of water)

The measurement of moisture content in the ducted air streams (evaporator inlet and outlet, and condenser outlet) was done using sampling systems. A manifold of perforated pipe was put into the air stream, and a small blower was used to draw a representative sample for the humidity measurement. The sample piping external to the duct was heavily insulated to reduce the influence from the ambient air. This piping configuration went through a number of revisions before a system was developed which had consistent performance.

The following section describes the instrumentation used at the key measurement points. Most of these have primary and secondary measurement systems to ensure accuracy. (The measurements on the condenser side of the system are in themselves a secondary method for determining system capacity.)

Evaporator Air Inlet

- Dry bulb Temperature:
  - Primary - 4 RTD probes with fast response tips inserted through the duct wall.
  - Secondary - a grid of 9 type “T” thermocouples.
- Humidity:
  - Primary - a General Eastern refrigerated mirror dew point sensor.
  - Secondary - dry and wet bulb temperature measurements of the sampled air stream.

Evaporator Air Outlet

- Dry bulb Temperature:
  - Primary - 4 RTD probes with fast response tips inserted through the duct wall.
  - Secondary - a grid of 9 type “T” thermocouples.
- Humidity:
  - Primary - a General Eastern refrigerated mirror dew point sensor.
  - Secondary - dry and wet bulb temperature measurements of the sampled air stream, and a Vaisala relative humidity sensor.

Condenser Air Inlet (measurements are samples of the room air near the test unit)

- Dry bulb Temperature:
  - Primary - 4 aspirated RTD probes
  - Secondary - 3 type “T” thermocouples, each with 6 junctions connected in parallel.
- Humidity:

– Primary - 4 wet bulb RTD probes downstream of the dry bulb RTDs.
– Secondary - 4 General Eastern relative humidity sensors.

Condenser Air Outlet

- Dry bulb Temperature:
  – 4 type “T” thermocouples, each with 3 junctions connected in parallel.
- Humidity:
  – Primary - dry and wet bulb temperature measurements of the sampled air stream
  – Secondary - a Vaisala relative humidity sensor.

Pressure Measurements

The measured pressures included the differential and upstream static pressures at the two air flow stations, and the external resistance to air flow for both the evaporator and condenser. Smart pressure transmitters were used for the measurements, and these were connected to manifolds that interconnected four pressure taps centered in each wall of the duct or flow station. Barometric pressure was also measured using a sensor located outside the rooms.

Electric Power Measurements

The total power demand of the system was measured using a Yokogawa 3-phase power meter, which also provided measured values of voltage, current, frequency, and reactive power. The demand of the subsystems including the compressor(s) and the evaporator and condenser fans were measured with watt transducers. Current transformers were used at all locations to isolate the instruments.

The instruments connected to the personal computer through different methods. The two dew point sensors connected directly to the computer through RS-232 serial cables, and would send a measured reading every second. The “slow response” sensors (mainly the RTDs and humidity sensors, but also some of the pressure transmitters and thermocouples) were connected to one of two small Hewlett Packard data loggers. These and the power meter communicated with the computer over a GPIB bus, and were polled for measurements every 10 seconds. The remainder of the mostly “fast response” sensors (pressure transmitters, thermocouples, watt transducers, and ambient weather sensors) were connected through a National Instruments signal conditioning system linked to a data acquisition card in the computer. This card was programmed to scan at a rate of 20 times a second, although ¼ second averages were used for some feedback control purposes and display updates, while 10 second averages were used for most displayed and logged results. The computer used a program written in National Instruments’ LabVIEW graphical programming language to acquire, process, display, and log the measured data, and to control the test unit and room conditioning systems.

Before installing the instruments in the ducts and unit testing, all of the temperature and pressure instruments were calibrated against laboratory standards through the data acquisition system. The calibration covered the range of conditions that each measurement was likely to experience. Correction curves were applied to all measurements to provide conformity with the standards. The TES standards lab calibrated all the power measurement systems, and the electronic humidity sensors were scaled using their as-delivered manufacturer’s calibration.

2.5 Test Results

The performance of these units is a function of several operating parameters, and it is not possible to give one performance result without specifying the specific operating conditions. ARI Standard 210/240-94 specifies a set of specific conditions for standard rating purposes. In addition, tests over a wider range of conditions allow creation of a set of performance curves for a unit, or generically for a technology. These curves could then be incorporated into energy simulation models to predict and compare the performance of these systems in actual buildings in various locations.

2.5.1 Baseline Unit (Test Unit #1)

The first unit tested was selected as representative of a typical rooftop system having the minimum rated efficiency for new construction. This unit is designated as the “baseline unit” as it will serve as the basis for comparison with later test units. As the first system to be installed in the new test facility, this unit was used to help optimize the measurement systems and the refine the operation of the facility. Therefore, the earliest test results were discarded due to subsequent changes in either instrumentation or procedure. While testing of this system actually began on May 12, the results presented here are from tests beginning on June 9 (after a dew point temperature sensor was added to the evaporator outlet sampling system). A summary of the presented test results is included in Appendix IV.

There were also two groups of tests performed on this unit. After the first group of tests was completed on June 16, the unit was modified with an evaporative pre-cooler. (The modified unit is designated as Test Unit #2, and the results from its tests are discussed later.) When the tests on the modified unit were completed, the pre-cooler was removed and an additional set of tests on the bare unit were conducted on July 23 and 26. Either as the result of instrument drift, or the addition of an extra layer of insulation around the evaporator discharge duct (added prior to the tests with the pre-cooler), the cooling capacity results from the second group of tests were as a whole slightly higher than from the first set. Post-test calibration checks revealed little drift in the primary instrumentation, so the slight shift was more likely due to improved measurement accuracy due to the extra insulation.

Table 5 lists the results obtained for the ARI Standard test conditions. Figure 2, Figure 3 and Figure 4 show the trends of cooling capacity, electric demand, and efficiency as a function of inlet air conditions. The ARI Standard test points are labeled on the figures.

Table 5. Results for Baseline Unit

ARI Test Designation	Inside T DB	Inside T WB	Outside T DB	Capacity (Tons)	Power (kW)	EER (Btu/Wh)	Flow (CFM)
A	80	67	95	7.40	11.1	8.02	3,320
B	80	67	82	8.03	10.4	9.28	3,330
C*	80	61	82	7.32	10.2	8.61	3,460
D*	80	61	82	1.37	2.2	7.49	3,550
Max	80	67	115	6.33	12.1	6.26	3,360
Low	67	57	67	7.23	9.1	9.50	3,330

*For Tests "C" and "D", the desired 57°F wet bulb temperature could not be achieved. Capacity shown for "C" and "D" is sensible cooling only. Test "D" is a cycling test in which the unit is on for 6 minutes and off for 24 minutes. Values of capacity and power for test "D" are given as energy usage normalized to one hour (i.e. Ton-hr / hr and kWh / hr).

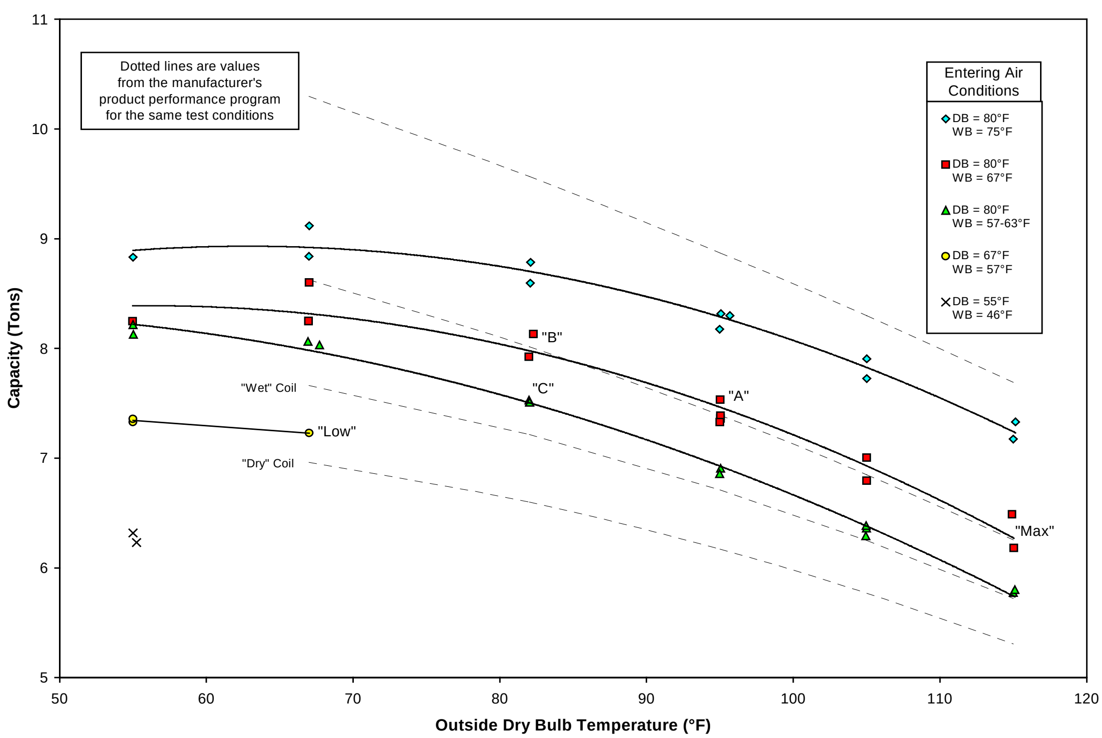

Figure 2. Baseline Unit – Cooling Capacity (Evaporator External Resistance: 0.25 WC)

The manufacturer of this test unit provided a performance predicting program intended for selecting a system for a particular application. This program was used with the actual test conditions for evaporator entering dry and wet bulb temperatures, outdoor dry bulb temperature, and evaporator air flow rate and resistance to produce curves of predicted performance. This performance represents how this particular model was designed to perform, and may not exactly predict the performance of this representative sample unit. These curves are included in the figures as dotted lines for comparison to the actual test results.

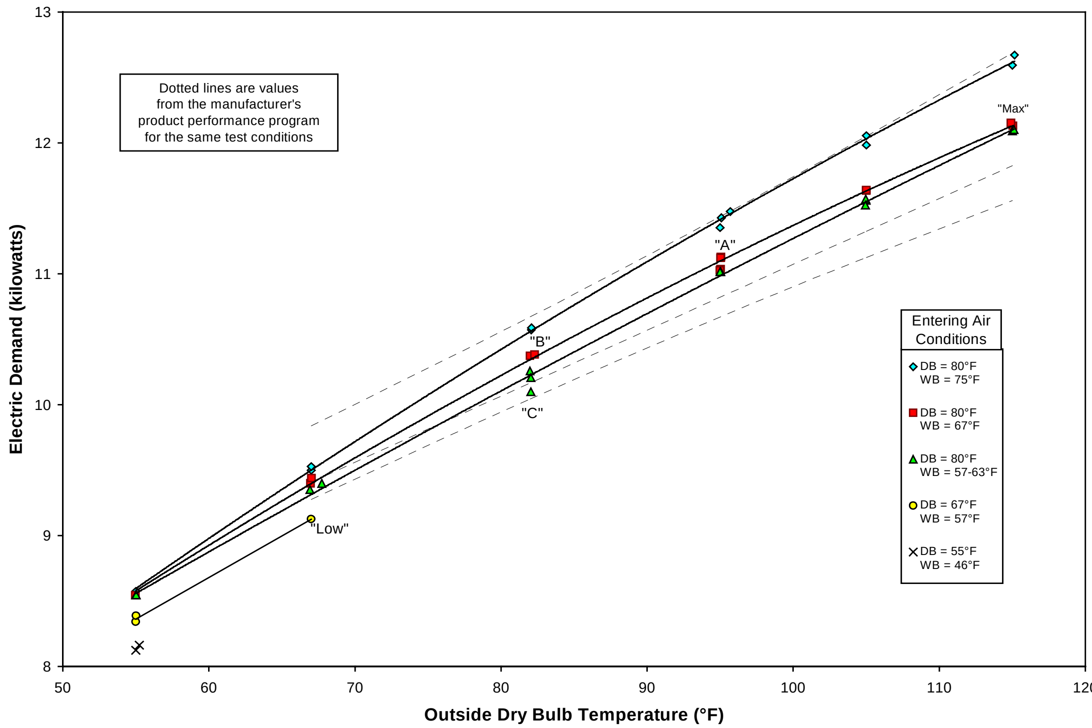

Figure 3. Baseline Unit – Total Electric Demand (Evaporator External Resistance: 0.25 WC)

One difficulty encountered in the operation of the test facility was achieving indoor room conditions that will result in no condensation on the evaporator - the "dry coil" test condition. This is required for ARI Standard tests "C" and "D", and particularly for test "D" which is a cycling test and requires the dry conditions so that the slow responding humidity instruments on the evaporator outlet may be ignored. To run a dry coil test at this facility, the ambient humidity ratio needs to be less than what is required for the test, which does not happen very often. The indoor room conditioning apparatus does not have the capability to cool or dehumidify; it can only add heat and humidity. For the steady-state "C" tests, it is possible to drive the humidity down by recirculating the return from the test unit, re-heating the discharge air that has been cooled and dehumidified. Theoretically, this recirculated air would eventually reach an equilibrium dew point equal to the temperature of the evaporator coil. The coil would be wet, but there would be no latent load and no condensate. However, some ambient air leaks into the room causing the humidity to creep up when the ambient humidity ratio is greater than the test conditions.

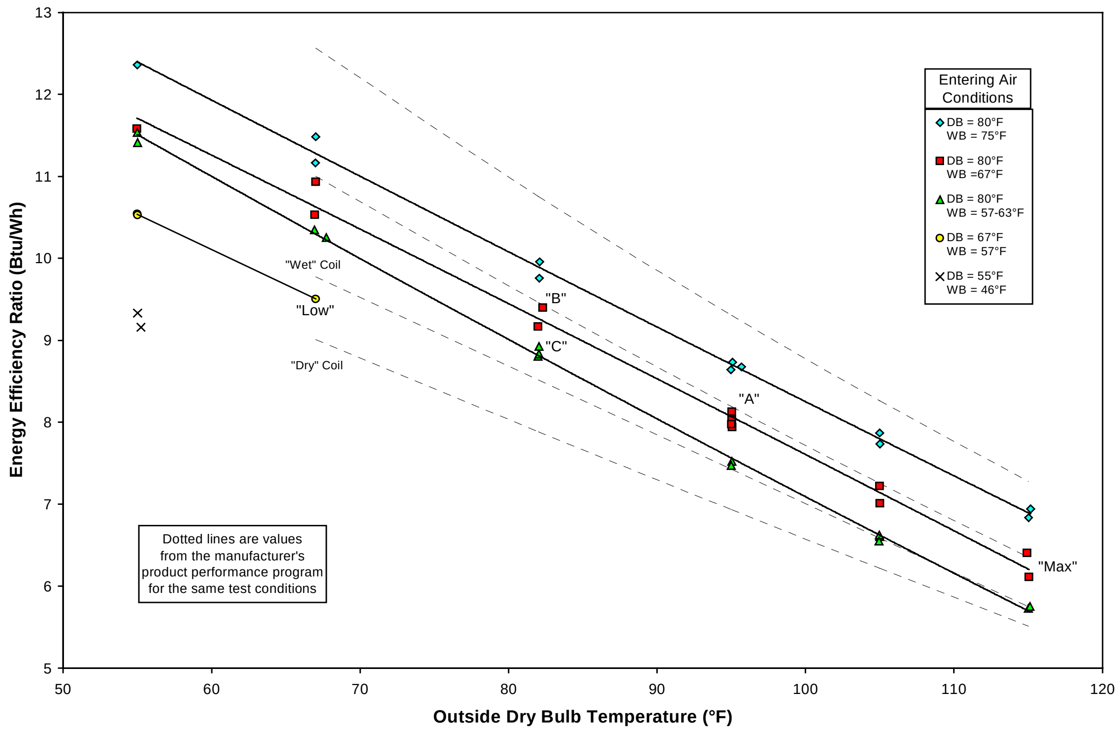

Figure 4. Baseline Unit – Energy Efficiency Ratio (Evaporator External Resistance: 0.25" WC)

For the cycling tests "D", the facility must be run with 100 percent outside air to maintain constant temperature and humidity conditions at the inlet while the output of the unit changes. This further restricts the ambient conditions under which the tests can be run to when the air has a humidity ratio and a dry bulb temperature less than what is required for the test. Since these conditions did not happen during the course of the testing, the result was that the tests that were supposed to be done with an indoor room condition of 80°F dry bulb and 57°F wet bulb were done at slightly higher humidity. The test wet bulb temperatures ranged from 59°F to almost 63°F. Dry coil tests were not even attempted in the second group of tests after the pre-cooler was removed because the ambient humidity was again too high.

The effect of the higher humidity was that the evaporator coil was most likely wet. Although the measured latent cooling load was minimal and there was no liquid water recovered at the drip pan, a damp coil has better heat transfer to the air resulting in a higher capacity. The manufacturer's performance program was used to investigate this, as the program determines from its input parameters whether the coil will be wet or dry. For each test condition input to the program, the entering wet bulb temperature was adjusted to the point where the coil transitions between dry and wet (a 0.1°F difference in wet bulb temperature), and the performance was recorded for both conditions. The end result is shown on the figures for capacity and efficiency as a pair of curves showing performance with a "dry coil", and performance with a barely “wet coil”. (Only one curve is shown in the electric demand figure as there was little difference.) The predicted “wet coil” curves are closer to the measured test results, which implies that the tests were probably done with a wet coil.

The capacity figure shows that the test results came very close to the capacity predicted by the performance program for the conditions of 80°F dry bulb and 67°F wet bulb, and reasonably close to the “wet coil” curve for the low humidity conditions. However, the measured capacity under high humidity (75°F wet bulb) was considerably less than the predicted values, with the difference increasing as the outside temperature decreases. The difference between the curves for the low humidity condition also diverge more as the outside temperature decreases, except that in this case there is more cooling provided than what is predicted. At the low outside air conditions, it appears as though the curves for the different levels of indoor humidity are converging towards some maximum capacity, whereas the model predicts that the curves should diverge as the temperature drops.

In the electric demand figure, the results predicted by the program are closest to the highest humidity test results, and it under-predicts the demand for the lower humidity cases. The end result in the figure for efficiency is that the measured results are bounded by the predicted values - higher at low humidity and lower as the humidity rises. Since the measured capacity improvement with decreasing outside temperatures was less than what was predicted, this unit may not show as much improvement with the evaporative pre-cooler than the performance program might indicate, lessening the economic benefit somewhat.

For most of the tests, the evaporator air flow resistance was held at a constant 0.25 inches of water, which is the minimum value for a unit of its rated capacity as defined in ARI Standard 210/240-94. However, this resulted in an airflow rate higher than the manufacturer’s rating conditions for this unit. It appears that in achieving the EER rating for this unit, the manufacturer may have used a different evaporator blower motor to achieve a low flow rate at the minimum resistance. This results in considerably less power consumption by the blower and may explain the published value of 9.7 kW instead of the measured 11.1 kW at nearly the same capacity, and the resultant energy efficiency ratio of 8.9 instead of the measured value of 8.0. This is particularly questionable since the manufacturer’s own performance program predicts performance at the rating conditions not much different from what was measured when the measured flow rate and resistance are entered into it. The average measured fan power through all the tests was 1.83 kW, which contributed from 14 to 24 percent of the total demand.

Figure 5 shows the sensitivity of various performance parameters to the evaporator external resistance, as it rises from the minimum of 0.25 inches of water. The results from this set of tests showed a general decrease in most parameters as the resistance is increased, although the EER peaked at a resistance of 0.5 inches (air flow rate at about 3,000 CFM) as the fan power consumption decreased faster than the capacity. This may explain why the published nominal air flow rate is near this value. However, the change in EER over the range of varied external static pressure was not great.

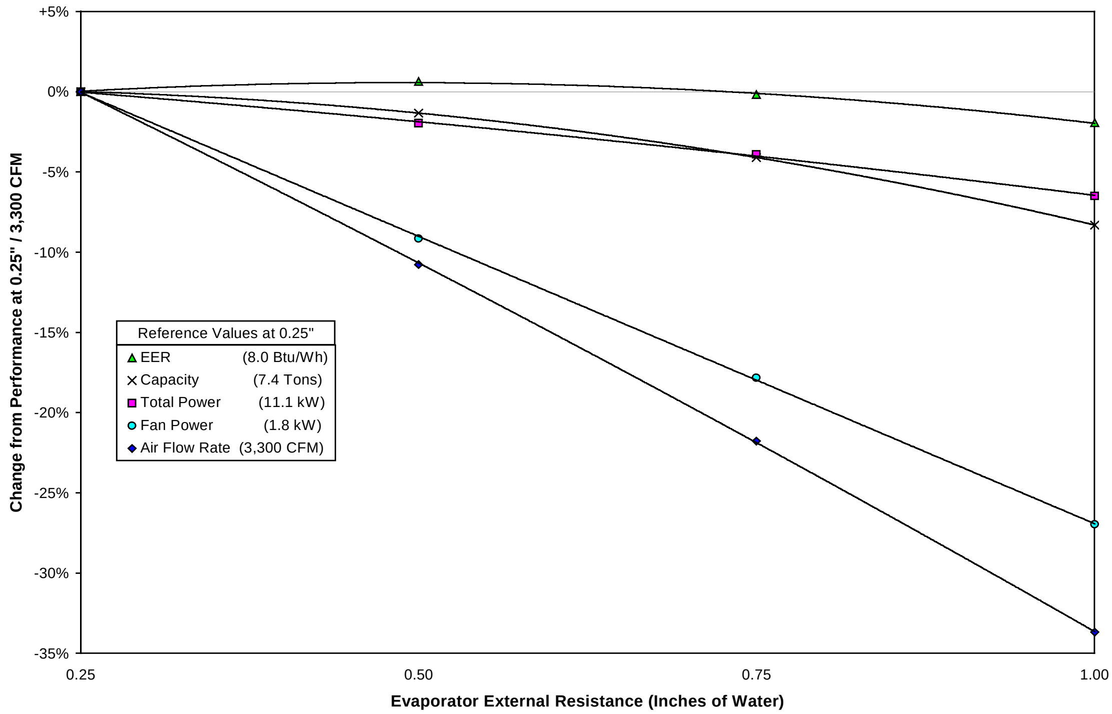

Figure 5. Baseline Unit – Effect of Evaporator External Resistance on Performance at Rating Conditions
(Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB)

Another performance parameter of interest is the coil bypass factor, which is a measure of the cooling coil performance. It is determined by assuming the air stream is made up of two components: one at the same temperature and humidity of the entering air, and the other at saturation at the evaporator coil temperature. The saturation point is found by extrapolating a line on a psychometric chart from the entering air conditions through the exiting air conditions until it intersects the saturation curve (100 percent relative humidity). This was done for most of the tests, and the results are included in Appendix IV. For some of the high humidity tests, the extrapolated line did not intersect the saturation curve, so the bypass factor is undefined. At the full load rating condition (Test "A"), the bypass factor averaged 28 percent, which is a bit high. Bypass factors are typically less than 20 percent.

As described in the test procedure, cycling performance tests may be used to determine a seasonal efficiency (SEER) that better describes the average performance of a system over a long time period. SEER values are normally only published for residential sized units (less than 5.4 Tons), but may be determined for other systems by the same method. Tests "B", "C" and "D" are used for the calculation. In addition to the cycling frequency prescribed by ARI for test "D" (6 minutes on and 24 minutes off, or a 30 minute cycle period with 20 percent on time) three other cycling frequencies were also examined to see what effect the frequency has on the end result. Since the cycling tests are prescribed to be dry coil tests, the moisture content at the evaporator outlet was assumed to be equal to that at the inlet and there is only sensible cooling done. As mentioned before, the coil was probably not perfectly dry, but the entering air was at as low a humidity as could be achieved. It was also necessary to circulate 100 percent outside air through both rooms of the test facility to maintain the desired stable conditions as the unit switches on and off. The following table gives the results from this set of tests, including calculated values of cooling load factor (CLF), degradation coefficient (C_D), and SEER:

Table 6. Results of Cyclic Performance Tests on the Baseline Unit

	Cycle Period (min.)	“On” Fraction	“On” Time (min.)	Sensible Capacity (Ton-hr/hr)	Electric Demand (kWh/h)	Sensible EER (Btu/Wh)	CLF	C_D	SEER
“C”	30	100%	30	7.32	10.2	8.61	100.0%		
	12	75%	9	5.47	7.83	8.39	74.8%	0.099	
	15	50%	7½	3.51	5.27	8.00	48.0%	0.135	
	20	25%	5	1.69	2.74	7.42	23.1%	0.180	
“D”	30	20%	6	1.37	2.20	7.49	18.8%	0.159	8.54

Figure 6 shows the measured test data from cycling test "D". This figure shows that the unit reaches equilibrium about 4-5 minutes after start. The degradation coefficient in the above table is related to how much of the total cycle on-time this transition period represents. The figure also shows that after the compressor has been signaled to shut off, the evaporator fan continues to operate to remove the residual cooling potential left in the evaporator coil. The values given in the table for capacity and power are integrated over a complete cycle period (30 minutes in this case). Thus the capacity includes this additional cooling provided after the compressor has shut off, and the power includes the unit standby power (mainly the demand of the compressor crankcase heater and controls).

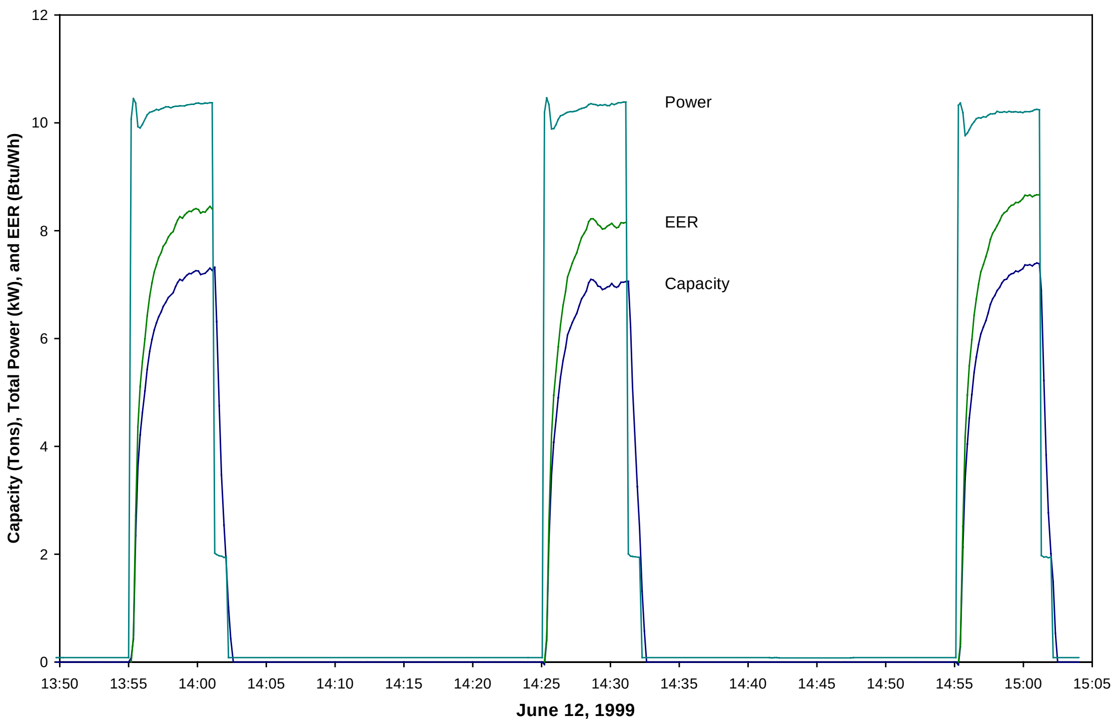

Figure 6. Baseline Unit – Cycling Performance
30 Minute Period, 20% on Fraction (ARI Standard Test "D")
(Evaporator Inlet: 80ºF DB, 61ºF WB (dry?); Condenser Inlet: 82ºF DB)

2.5.2 Baseline Unit with Evaporative Pre-Cooler on the Condenser (Test Unit #2)

For the second test unit, the baseline unit was modified with a commercially available evaporative pre-cooler to condition the intake condenser air. Water from a pan is distributed over a cellulose evaporative media by a small pump. The evaporative media acts to increase the surface area for evaporation and reduce spray carry-over. The pump operates whenever the compressor is active and the ambient temperature is above 75°F. A tap in the pump discharge line directs some of the flow to a drain to reduce the buildup of salts in the pan.

The addition of the evaporative media adds some restriction to the flow of air through the condenser. This results in a decrease in airflow rate, which without evaporation would cause an increase in condensing temperature and pressure and compressor power. The water pump also consumes energy, and there is a minor cost involved in the water consumed by the system. Thus, the goal of this test phase is to determine under what outside conditions the pre-cooler is a cost-effective addition.

The pre-cooler used was intended for a unit with a larger condenser face area than the baseline unit. In order to be fair, a duct was constructed with the same face area as the baseline unit to connect to the pre-cooler, and the airflow through the evaporative media outside of this area was blocked. If this had not been done, the air velocity through the larger area would have been reduced, resulting in a lower flow restriction and more evaporation, and ultimately better system performance.

The tests on this system were conducted between June 24 and July 22. Since the testing of this system added another variable because of the effect of outside room humidity, testing under only one indoor room condition reduced the number of tests. This was the same condition as for most of the standard rating tests: a dry bulb temperature of 80°F and a wet bulb temperature of 67°F. For consistency with the baseline unit tests, the evaporator airflow resistance was also kept at 0.25 inches of water column. The only deviations from these conditions were for a few cyclic tests that require a lower wet bulb temperature to the evaporator (dry coil).

Prior to testing, a matrix was made from four outdoor room dry bulb temperatures, with various levels of humidity provided by choosing two to three wet bulb temperatures for each. Since the pre-cooler is not designed to operate below 75°F, the dry bulb temperatures were taken from the values used in the baseline tests that were above this level (Table 7).

Table 7. Outside Room Test Condition Matrix for the Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit

	Dry bulb Temperatures (°F)							
	82		95		105		115	
Wet bulb	60	(26%)	65	(18%)	70	(16%)	75	(15%)
Temperatures	70	(55%)	75	(40%)	80	(34%)	85	(30%)
°F (%RH)			85	(67%)	90	(56%)	95	(48%)

The ARI Standard Test Procedure gives values for outdoor room wet bulb temperature for rating evaporative systems. Test point “A” has the conditions of 95°F dry bulb and 75°F wet bulb, which was included in the matrix of tests. The maximum operating condition test was also included with 115°F dry bulb and 75°F wet bulb. However, the outdoor wet bulb condition for test point “B” was overlooked; it was actually bounded by the two values chosen. (Test “B” is 82°F dry bulb with 65°F wet bulb.) In retrospect, it would have been better to select the wet bulb temperatures based on levels of relative humidity.

Table 8 lists the test results for the standard test conditions along with the change from the baseline tests. Deviations from the rating temperature conditions of more than 1°F are indicated by listing the actual test condition below the desired condition. The values shown for test “B” have been interpolated from the test results for the required wet bulb condition. The table shows similar improvements in performance in Standard tests “A” and “B”, which is most likely the result of these tests having about the same relative humidity (about 40 percent). The results for the maximum condition test show significant improvement due to this test being conducted at a very low relative humidity (15 percent). The power consumption of the pre-cooler water pump was measured and added to the measured total of the unit. However, it is an insignificant fraction of the total at only 111 Watts (~one percent).

Table 8. Results for the Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit

(Percent Change from Baseline Unit)

ARI Test Designation	Inside T DB	Inside T WB	Outside T DB	Outside T WB	Capacity (Tons)	Power (kW)	EER (Btu/Wh)	Evap. Flow (CFM)	Cond. Flow (CFM)
A	80	67	95	75	7.69	10.7	8.58	3,320	4,780
					+3.8%	-3.0%	+7.0%	-0.1%	-7.9%
B	80	67	82	65	8.30	10.1	9.90	3,320	4,820
				(Interpolated)	+3.4%	-3.1%	+6.7%	-0.3%	-6.2%
C	80	57	82	65	7.64	9.9	9.28	3,460	4,790
		(61)		(63)	+4.4%	-3.2%	+7.8%	0.0%	-9.4%
D	80	57	82	65	1.46	2.1	8.30	3,570	4,770
		(62)	(83)		+6.3%	-4.1%	+10.8%	+0.5%	-9.0%
Max	80	67	115	75	7.30	11.3	7.78	3,350	4,720
					+15.2%	-7.3%	+24.3%	-0.5%	-7.8%

Test “D” is a cycling test in which the unit is on for 6 minutes and off for 24 minutes.
Values of capacity and power for test “D” are given as energy usage normalized to one hour
(i.e. Ton-hr / hr and kWh / hr).

The performance of an evaporative air cooler is measured by its effectiveness, or the ratio of the measured reduction in dry bulb temperature across the pre-cooler to the difference between the outside dry and wet bulb temperatures (wet bulb depression). The outlet temperature of the pre-cooler is measured using a thermocouple grid made up of 18 junctions arranged across the condenser inlet combined into three measurement points. Despite the relatively large number of sensing points, it may not represent the true average value of the air temperature. However, calculations based on this measurement and the measurements of outside dry and wet bulb temperature produced a fairly constant result over the range of test conditions. The average measured pre-cooler effectiveness was 51 percent, with a range of 46 percent to 56 percent. The manufacturer's literature implies an effectiveness of about 60 percent, and the lower test result may have to do with the modifications made to make it work with this test unit. The evaporative pad area for this test unit was approximately 1.5 ft^2/ton, at the low end of the range recommended by the manufacturer (1.5 - 2 ft^2/ton). The evaporative media was examined for the extent of wetting during several tests, and there were only a few dry streaks visible.

The performance of Test Unit #2 is shown in the following several figures. The data contained in the figures as well as other test results are included in Appendix IV. The first set of two figures shows different views of the measured cooling capacity. Figure 7 shows the results as a function of the outdoor room dry bulb temperature, for direct comparison with the figures for the baseline unit. The various humidity conditions are indicated by grouping the data into bands of relative humidity. A bi-quadratic curve fit of the capacity as a function of the dry bulb temperature and relative humidity was calculated, and the trends of the midpoint relative humidity for each group have been drawn. Also, the performance trend for the baseline unit has been added for reference. The figure shows that the capacity improved as outside humidity decreased, although at high humidity when there is little temperature reduction from evaporation, the capacity was actually worse than the baseline unit.

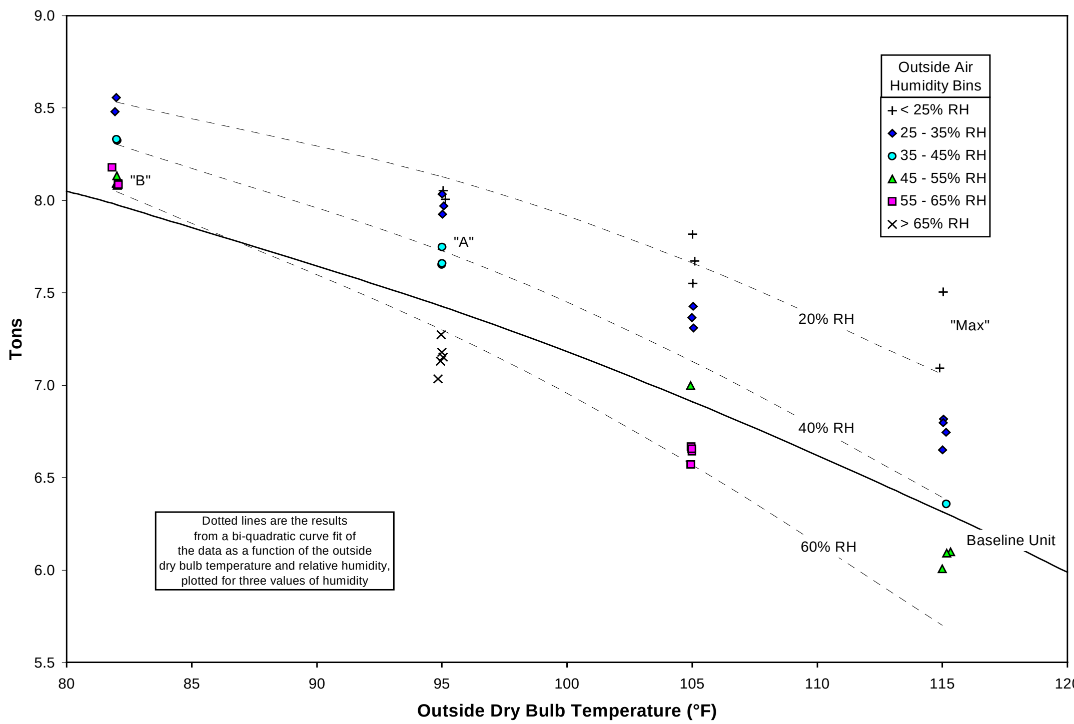

Figure 7. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Cooling Capacity versus Outside Dry bulb Temperature and Relative Humidity (Evaporator Inlet: 80ºF DB, 61ºF WB (dry?); Condenser Inlet: 82ºF DB)

Figure 8 shows the same data, but arranged differently. This time the capacity is graphed as a function of the relative humidity and grouped by the dry bulb temperature. The results for the baseline unit are included as dashed horizontal lines, since it should not have been affected by the outside relative humidity. Added to the figure is a curve that connects the intersections of the data trends with the baseline values, indicating the relative humidity at which there is no improvement in performance. The resultant curve shows that as the outside dry bulb temperature increases, the relative humidity must decrease for there to be an improvement in performance.

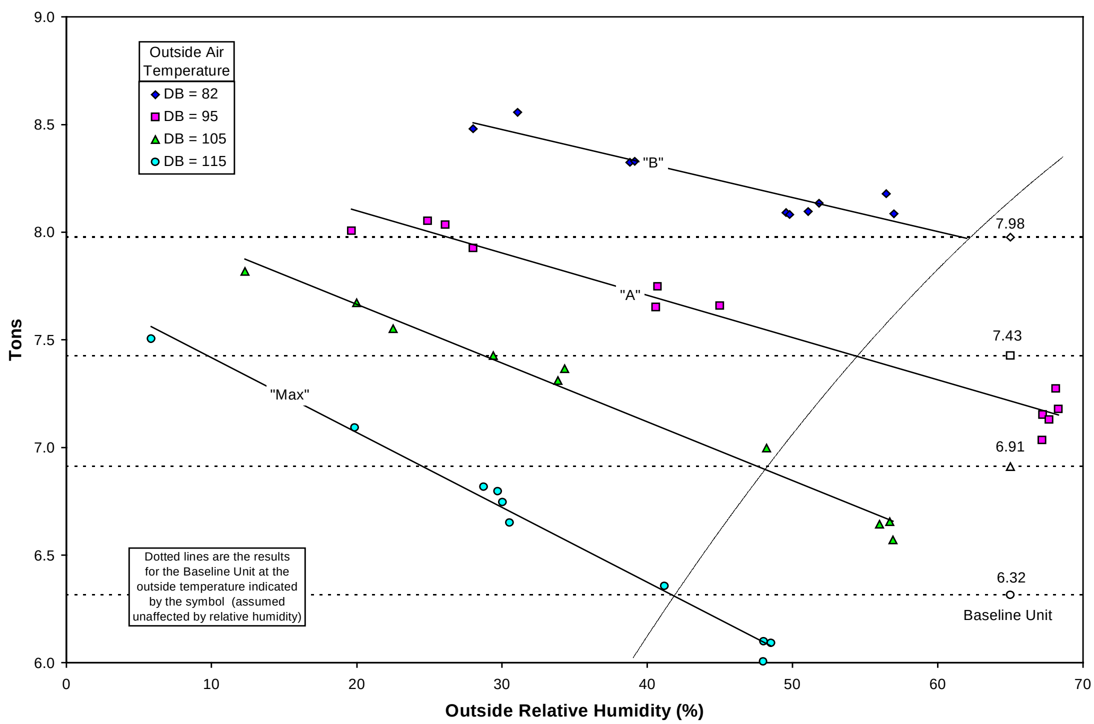

Figure 8. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Cooling Capacity versus Relative Humidity and Outside Dry bulb Temperature (Evaporator Inlet: 80ºF DB, 67ºF WB; Evaporator External Resistance: 0.25” WC)

The next pair of figures (Figure 9 and Figure 10) shows the same relationships for the total power demand of the test unit. Again, bi-quadratic curve fit of the data has been calculated, and trends have been added representing the midpoints of the groups. In all of the tests, the demand was less than that for the baseline unit over the tested dry and wet bulb temperatures. However, when viewed as a function of the pre-cooler outlet temperature, the demand actually increased by about two percent over that of the baseline unit for the same temperature.

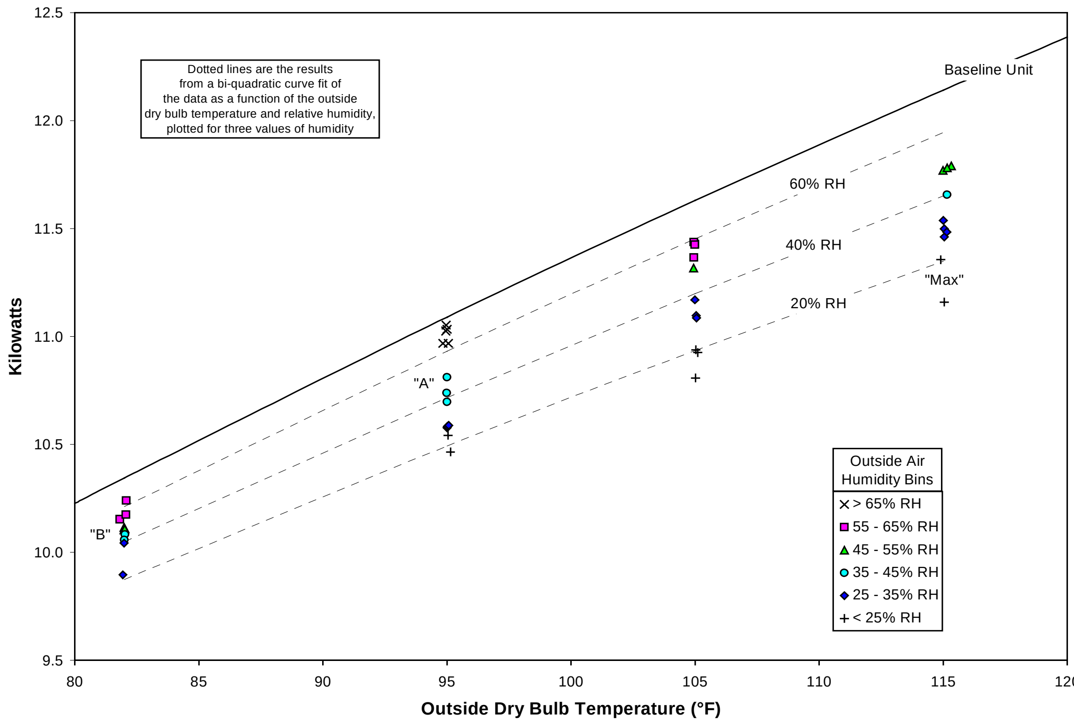

Figure 9. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Total Electric Demand versus Outside Dry bulb Temperature and Relative Humidity (Evaporator Inlet: 80ºF DB, 67ºF WB; Evaporator External Resistance: 0.25” WC)

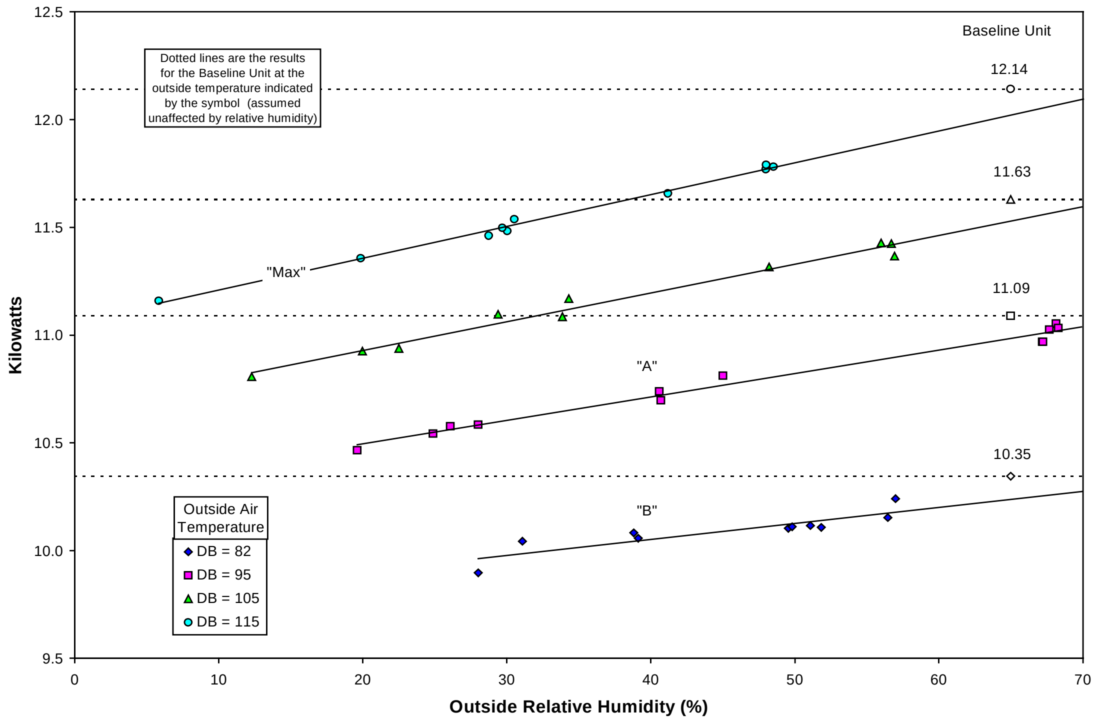

Figure 10. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Total Electric Demand versus Relative Humidity and Outside Dry bulb Temperature

(Evaporator Inlet: 80ºF DB, 67ºF WB; Evaporator External Resistance: 0.25” WC)

The final set of figures in this groups (Figure 11 and Figure 12) show the results for the energy efficiency ratio. With increases in capacity and reduction of demand, the efficiency is improved considerably. With the overall reduction in demand, the minimum outdoor relative humidity necessary for improved performance has increased from that shown for the cooling capacity alone.

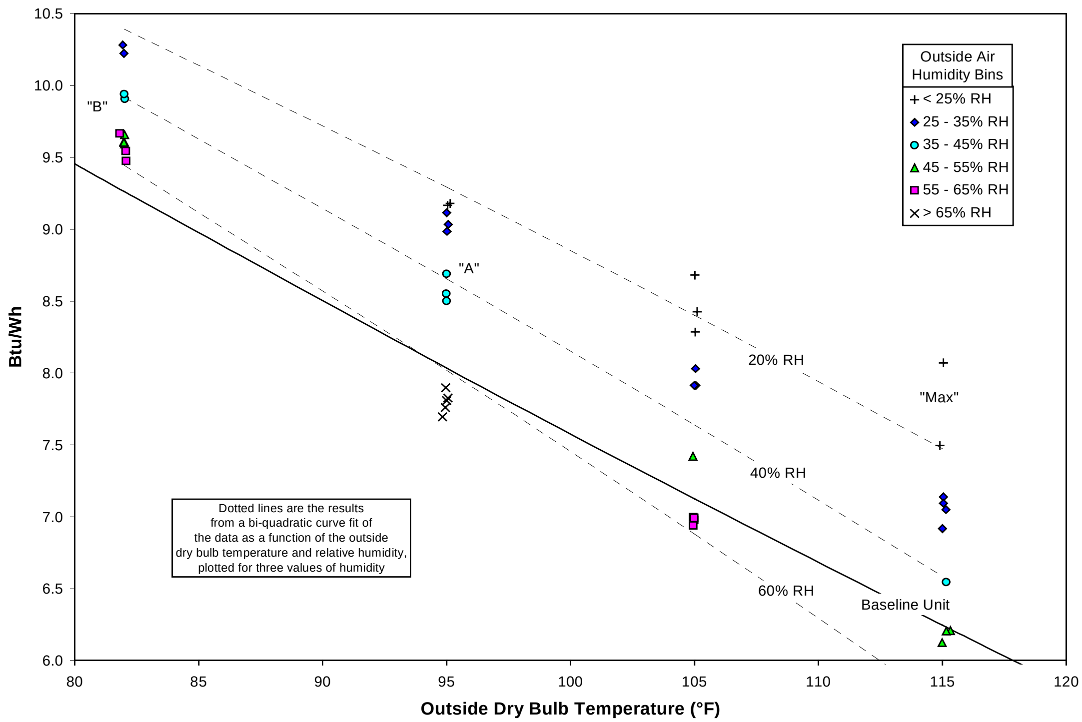

Figure 11. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Energy Efficiency Ratio versus Outside Dry bulb Temperature and Relative Humidity
(Evaporator Inlet: 80ºF DB, 67ºF WB; Evaporator External Resistance: 0.25” WC)

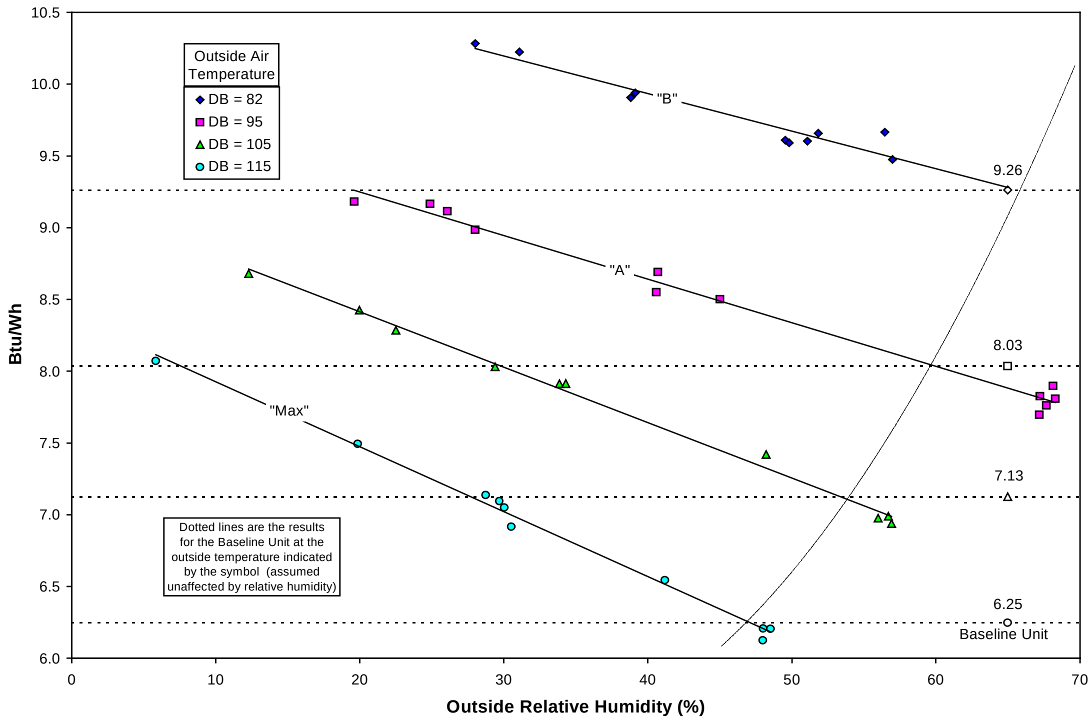

Figure 12. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Energy Efficiency Ratio versus Relative Humidity and Outside Dry bulb Temperature

(Evaporator Inlet: 80ºF DB, 67ºF WB; Evaporator External Resistance: 0.25” WC)

Figure 13 shows the water usage rate for the pre-cooler. This chart shows that the water consumption is primarily related to the outside wet bulb depression, with little effect from the dry bulb alone. The "intercept" of the trend line (where the wet bulb depression is zero and there should be no evaporation) is an indicator of the rate of flow being diverted to the drain to maintain water quality. Even at low humidity (as represented by a high wet bulb depression) the water usage rate is still very low at less than 16 gallons per hour (0.27 GPM).

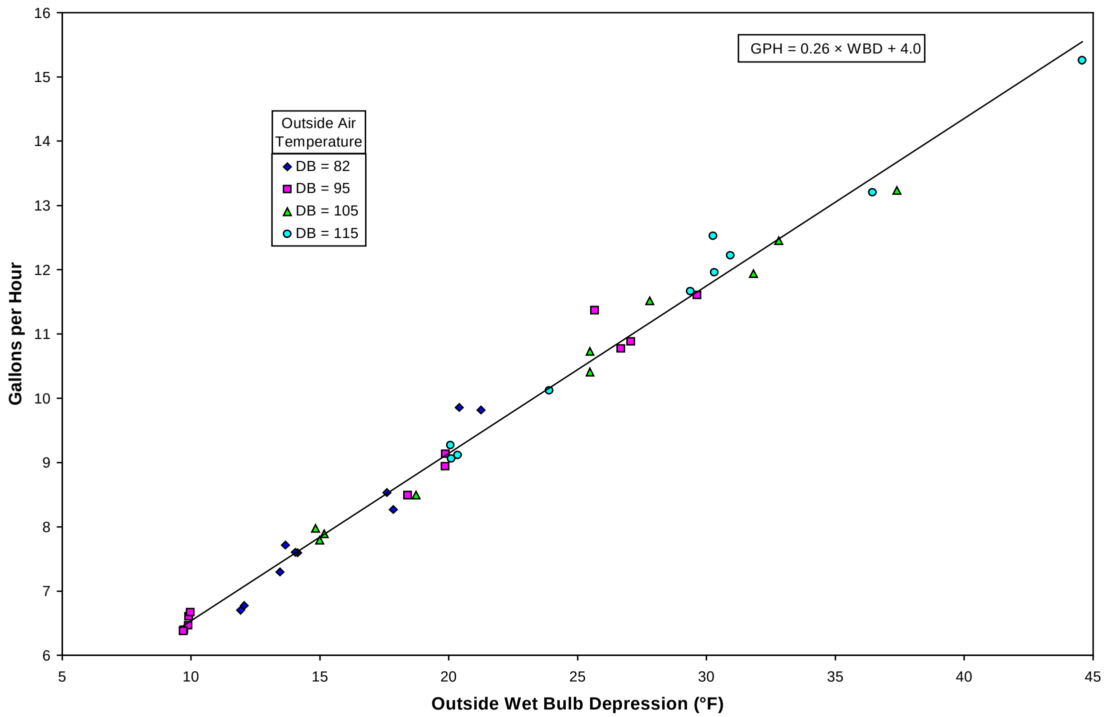

Figure 13. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Water Usage Rate

The external resistance on the condenser is maintained through the control of a variable speed booster fan downstream of the airflow measurement station. Figure 14 shows the sensitivity of various performance parameters to the condenser external resistance, as it decreases from the normal value of zero inches of water. The decrease in pressure causes an increase in airflow rate through the condenser. At a resistance of -0.13 inches, the airflow rate has been increased to about the level of the baseline unit without the pre-cooler, and this point is included in the chart. The results show the airflow rate and system efficiency increase with decreasing resistance, and the power demand (both total and condenser fan) and pre-cooler effectiveness decreased. Capacity stayed about the same.

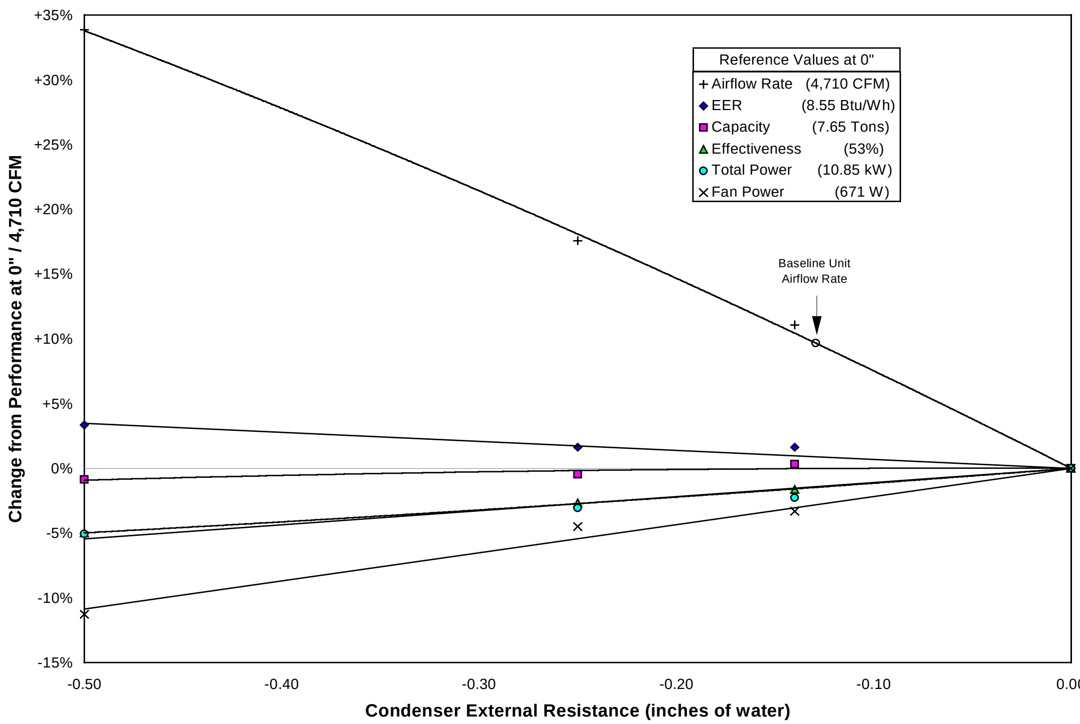

Figure 14. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Effect of Condenser External Resistance on Performance at Rating Conditions
(Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB, 75ºF WB)

A set of cyclic performance tests was performed on this system similar to those done on the baseline unit. The pre-cooler pump was controlled to turn on and off along with the system compressor. The longer the unit is off, the more potential there is for the pre-cooler pad to dry out, leading to less effective performance when the water is restarted.

Table 9 gives the results from these tests, including calculated values of cooling load factor (CLF), degradation coefficient (C_D), and SEER:

Table 9. Cyclic Performance Test Results on Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit

	Cycle Period (min.)	"On" Fraction	"On" Time (min.)	Sensible Capacity (Ton-hr/hr)	Electric Demand (kWh/h)	Sensible EER (Btu/Wh)	CLF	C_D	SEER
"C"	30	100%	30	7.64	9.88	9.28	100.0%		
	12	75%	9	5.84	7.63	9.19	76.4%	0.042	
	15	50%	7½	3.81	5.05	9.07	49.9%	0.045	
	20	25%	5	1.86	2.64	8.46	24.4%	0.117	
"D"	30	20%	6	1.46	2.11	8.30	19.1%	0.130	9.26

Figure 15 shows the measured test data from cycling test "D". In comparison with the results for the baseline unit, this figure shows that the unit takes longer to reach equilibrium in capacity as the pre-cooler reaches a stable level of wetness.

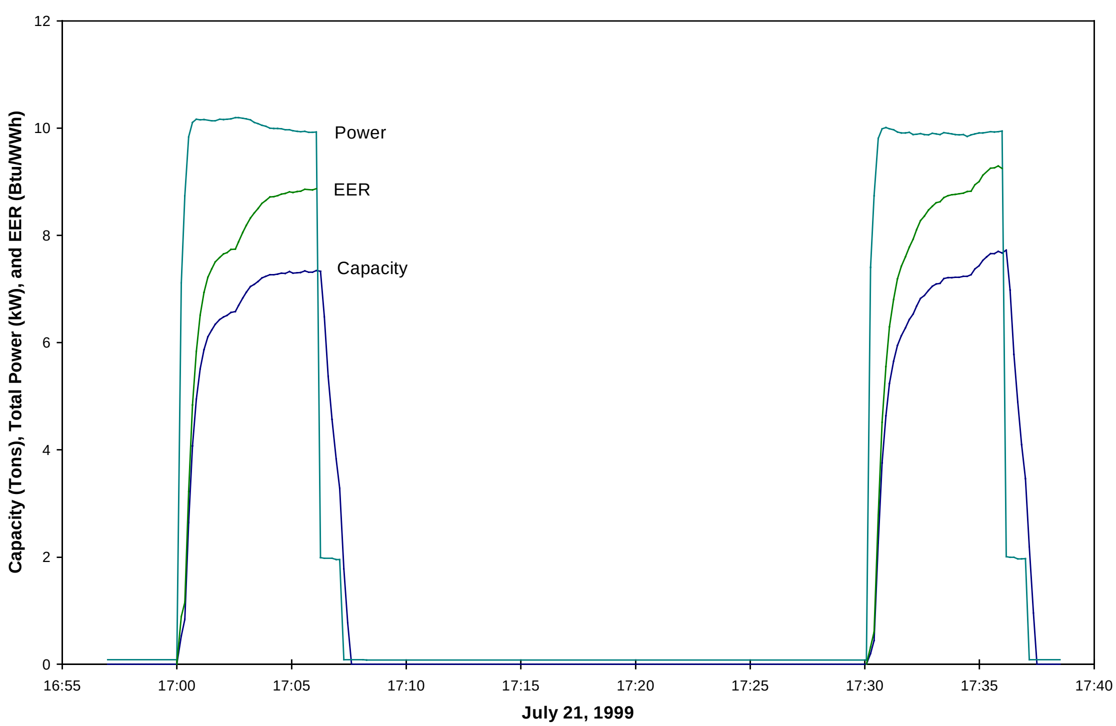

Figure 15. Baseline Unit with Evaporative Pre-Cooler on the Condenser Unit – Cycling Performance 30 Minute Period, 20% on Fraction (ARI Standard Test "D") (Evaporator Inlet: 80ºF DB, 61ºF WB (dry?); Condenser Inlet: 82ºF DB)

2.5.3 High Efficiency Dual Compressor (Test Unit #3)

The third and final unit tested in this group was selected as a high efficiency unit using two compressors. The concept behind dual compressors is that the unit can be staged to provide a lower cooling capacity when required rather than cycling the entire unit on and off. This reduces wear on the components from fewer starts, and avoids the performance penalty as the unit approaches stability. The dual compressor system also allows for cost savings from being able to install smaller compressors, which is somewhat offset through selecting higher efficiency models. Higher efficiency units also tend to have greater condenser face area for better heat transfer and reduced resistance to flow.

For most of the sensitivity tests on this unit, it was run at full load with both compressors operating. While checking the performance after the initial startup of the unit, the performance was not as expected. The cooling capacity was lower than specified, and the power consumption was higher, resulting in a low efficiency. The consensus was that either some of the refrigerant charge may have leaked, or it may be contaminated with other gases. To address the first theory, refrigerant charge was added to the two refrigerant circuits to bring suction pressures and superheat to expected levels. Capacity increased, but power also increased to a point where performance was still very low. At this point, the refrigerant was evacuated from both loops, which were then recharged with new refrigerant. This did provide some improvement in performance, but it was still not up to the design specification.

Further study of the specifications in comparison with the measured values suggested that the evaporator fan in the test unit was larger than the one used for rating purposes, or the rated unit had a special drive to reduce RPM at lower static pressures. Other evidence to support this was that at the minimum external resistance of 0.25 inches of water, the airflow rate was much greater than the design value. To obtain the rated evaporator airflow rate of about 3,000 CFM, the external resistance had to be set to 0.7 inches. This resistance value was maintained through most of the testing, although the airflow rate did fluctuate slightly (±4 percent range) around the average value mainly from changes in density. In order to better compare the manufacturer's specifications with the test results, the specifications were adjusted for the average measured values for both the evaporator and condenser fan power (2.30 and 0.62 kilowatts, respectively). The adjusted specifications are included in the following figures. The average evaporator fan power of 2.30 kW made up between 19 and 33 percent of the total power demand for this unit.

As was done for the previous units, the results obtained for the ARI Standard test conditions are listed in Table 10, and the trends of cooling capacity, electric demand, and efficiency as a function of inlet air conditions are shown in Figure 16, Figure 17 and Figure 18. The ARI Standard test points are labeled on the figures. The table contains more tests because of the different modes of operation created by having two compressors.

Table 10. Results for the High Efficiency Dual Compressor Unit

Subscripts: 2 - Both compressors running, 1 - One compressor only

ARI Test Designation	Inside T DB	Inside T WB	Outside T DB	Capacity (Tons)	Power (kW)	EER (Btu/Wh)	Flow (CFM)
A2	80	67	95	6.91	9.9	8.40	2,860
A1	80	67	95	3.16	6.4	5.97	2,900
B2	80	67	82	7.05	8.9	9.54	2,850
B1	80	67	82	3.31	5.9	6.78	2,900
C2*	80	62	82	6.11	8.8	8.34	2,880
D2*	80	62	82	3.47	6.5	6.45	2,950
C21*	80	57	82	6.58	8.9	8.89	3,020
D21*	80	57	82	1.21	1.8	7.88	3,030
C1*	80	61	82	2.91	5.9	5.92	2,970
D1*	80	60	82	0.56	1.2	5.39	2,990
Max2	80	67	115	6.24	11.8	6.34	2,910
Low2	67	57	67	6.13	8.0	9.21	2,970
IPLV2	80	67	80	6.93	8.8	9.44	2,910
IPLV1	80	67	80	3.32	5.8	6.86	2,910

* For Tests “C” and “D”, the desired 57°F wet bulb temperature could not always be achieved. Capacity shown for these tests is sensible cooling only. Test “D” is a cycling test in which the unit is on for 6 minutes and off for 24 minutes. The numbers following the “D” indicate which compressors are cycled (i.e. 2: compressor 2 cycling, compressor 1 on always, 21: both compressors cycling, 1: compressor 1 cycling, compressor 2 off always). The “C” test with the same number is the steady-state test that immediately preceded the cycling test. Values of capacity and power for test “D” are given as energy usage normalized to one hour (i.e. Ton-hr / hr and kWh / hr).

The performance of this unit was significantly different than that of the baseline unit, particularly at low outside temperatures. Although adjusted for the actual measured values for fan power, the test results are low even in comparison with the manufacturer’s own specifications, and do not follow the same trends. In the baseline unit, the capacity was lowest at highest outside temperatures, and improved as the outside temperature decreased. The capacity is not a linear function with outside temperature, and it begins to level out as the temperature reaches a minimum. In this unit, a maximum capacity is reached at a relatively high outside temperature, and then begins to decrease as the temperature continues lower. The maximum capacity and the outside temperature at which it is reached are also dependent on the entering wet bulb temperature (which affects the evaporator temperature and pressure).

The reason for the capacity reduction at low temperatures may be the result of a reduction in the refrigerant flow. This unit uses fixed orifices (capillary tubes) to cause the pressure drop to the evaporator. As the outside temperature decreases, the pressure of the condensing refrigerant also decreases. Thus, there will be less force to push the refrigerant through the orifices, resulting in less flow. An expansion valve that can adjust to the changes in condenser temperature and pressure to allow more flow would probably have been a better option if more cooling is required under these conditions. However, if the outside temperature gets this cool, it becomes more economical to cool a space through ventilation.

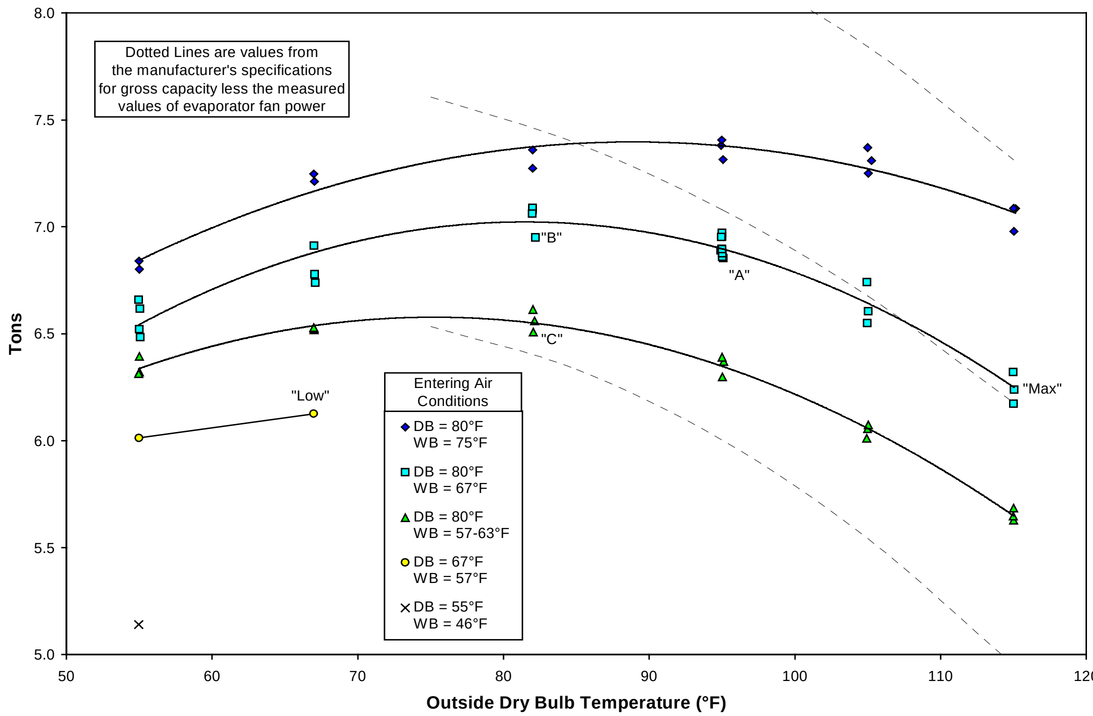

Figure 16. High Efficiency Dual Compressor Unit – Cooling Capacity (Evaporator External Resistance: 0.70” WC)

As for the cooling capacity at the rating conditions (ARI Standard Test "A"), the results are actually not that much different from the adjusted specifications. The average test result for these conditions was a capacity of 6.91 tons, versus an adjusted specification of 7.06 tons - a difference of only 2.4 percent. At higher outside temperatures at the same evaporator inlet conditions, the results are even closer to the specifications. The areas where it differs the most are in tests with high evaporator inlet humidity (except at the very highest outside temperature), tests with the rated evaporator inlet humidity at outside temperatures below the standard condition, and for most dry coil tests, especially at the higher temperatures. The shape of the specification curve for the low humidity case is similar to the two at higher humidity, but it is shifted down from the test results. The difference may be due to the same reason as for the baseline unit: the specifications may be for a completely dry coil, whereas the test results are probably with a wet coil.

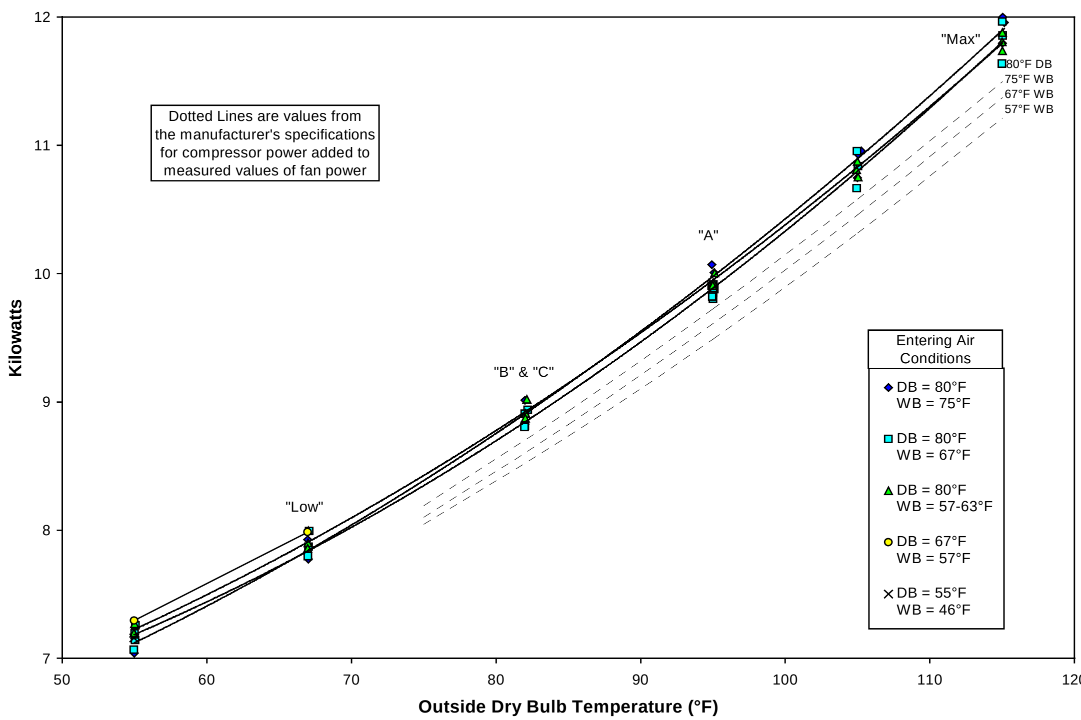

Figure 17. High Efficiency Dual Compressor Unit – Total Electric Demand (Evaporator External Resistance: 0.70" WC)

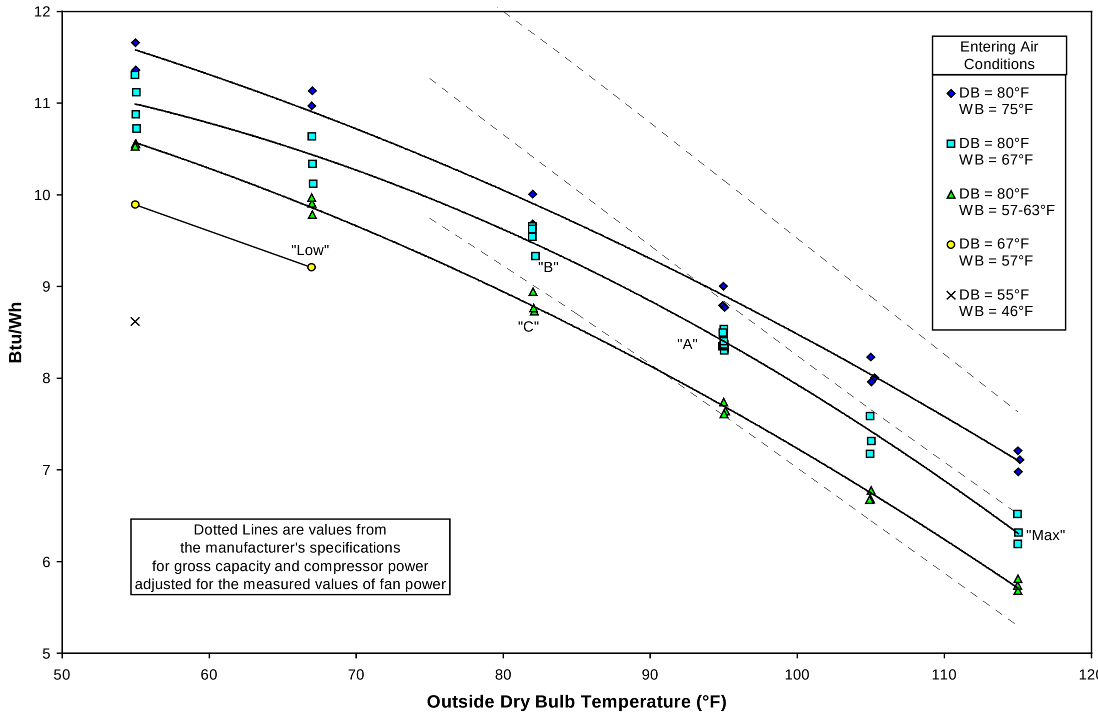

Figure 18. High Efficiency Dual Compressor Unit – Energy Efficiency Ratio (Evaporator External Resistance: 0.70" WC)

The figures show a noticeable spread in the capacity results for the same test conditions. Some of this is due to instabilities in the test conditions, particularly in the tests with a high evaporator inlet humidity since the indoor room humidifier could not produce a constant output. Extra testing at the standard rating conditions showed a slight sensitivity to outdoor room relative humidity, with higher outside room humidity resulting in a lower capacity. As this unit did not involve evaporation on the condenser side, the outdoor humidity was not controlled to a particular value and was allowed to vary with the ambient humidity level. It was first thought that the difference in humidity might be affecting the heat transfer at the condenser. However, raising the humidity of air at a constant temperature results in a decrease in density, but an even greater increase in the specific heat. Thus, at a constant volumetric airflow rate, the heat capacitance rate (multiple of the volumetric flow, density, and specific heat) would rise, resulting in an increased heat transfer capability that should actually give an improvement in overall performance.

The decrease in capacity seen at higher outside humidity is probably due to some air leakage from the outdoor room to the evaporator fan intake. While most rooftop units are designed to take in a fixed volume of outside air for ventilation, it was assumed that the test unit had all of the routes for air intake sealed. There are not any apparent intakes external to the unit, so it is unknown how the outside air could be leaking in. Figure 19 shows the results of the tests on the sensitivity to outdoor room humidity. It shows that while the capacity (as measured through the evaporator) and the resulting efficiency decrease as the humidity rises, the power consumption, evaporator and condenser airflow rates, and the capacity as measured on the condenser side stay relatively constant.

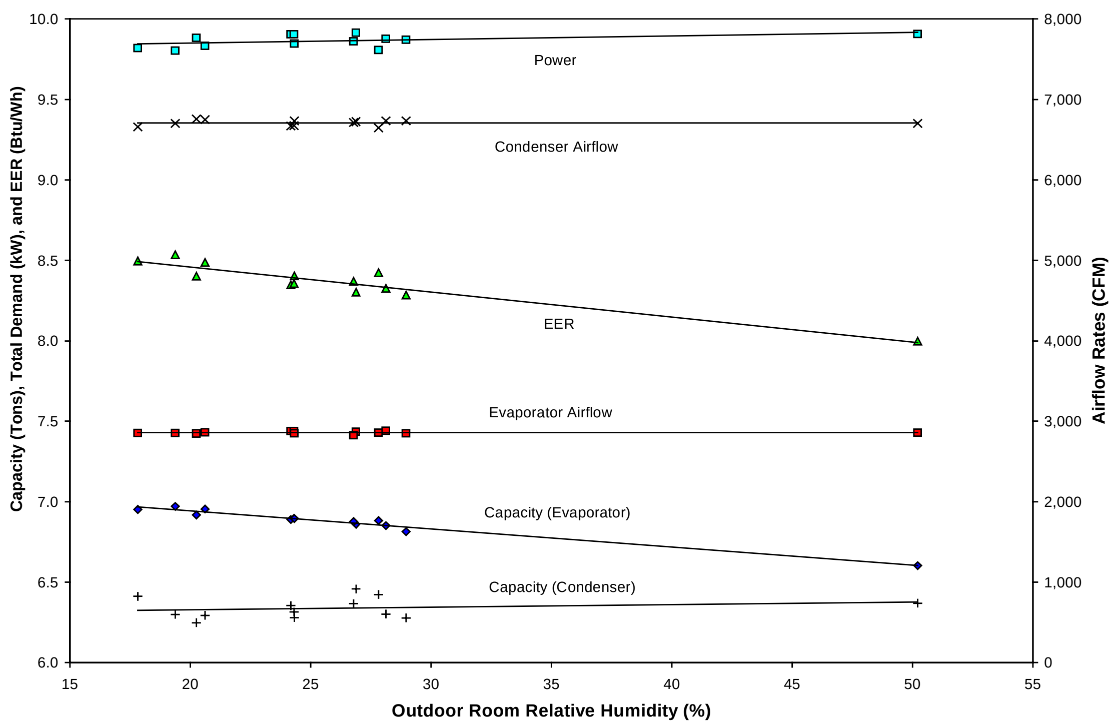

Figure 19. High Efficiency Dual Compressor Unit – Sensitivity to Outdoor Room Humidity (Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB)

The figure for total electric demand shows a consistent decrease with decreasing outside temperatures over the entire range of test conditions. This unit also has less sensitivity to the evaporator inlet humidity than did the baseline unit. The manufacturer's specifications have the same shaped curves as the test results but with a bit more sensitivity to the inlet humidity (although still not as much as the baseline unit). Overall the measured power demand was greater than the specifications, but not by much.

The large decrease in power consumption acted to offset the leveling out and reduction in capacity at lower temperatures to produce curves of efficiency that continue to rise as outside temperature drops. However, the curves do deviate significantly from the manufacturer's specifications under the low temperature conditions. At the standard rating conditions, the resulting average EER of 8.40 was below the specification of 8.83, but this is only a difference of

4.9 percent. Compared with the baseline unit, the EER of this unit was 4.7 percent greater at the standard rating conditions.

Tests were also performed to examine the sensitivity of the performance to both the evaporator and condenser external resistance as they deviate from their test conditions (Figure 20 and Figure 21).

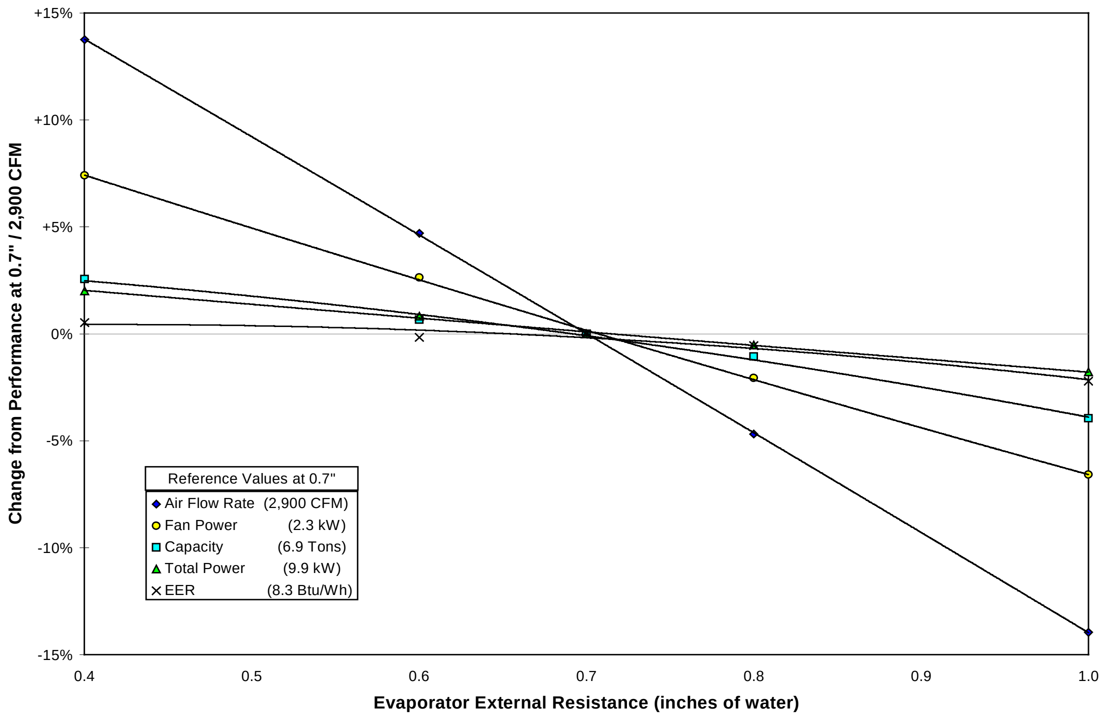

Figure 20. High Efficiency Dual Compressor Unit – Effect of Evaporator External Resistance (Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB)

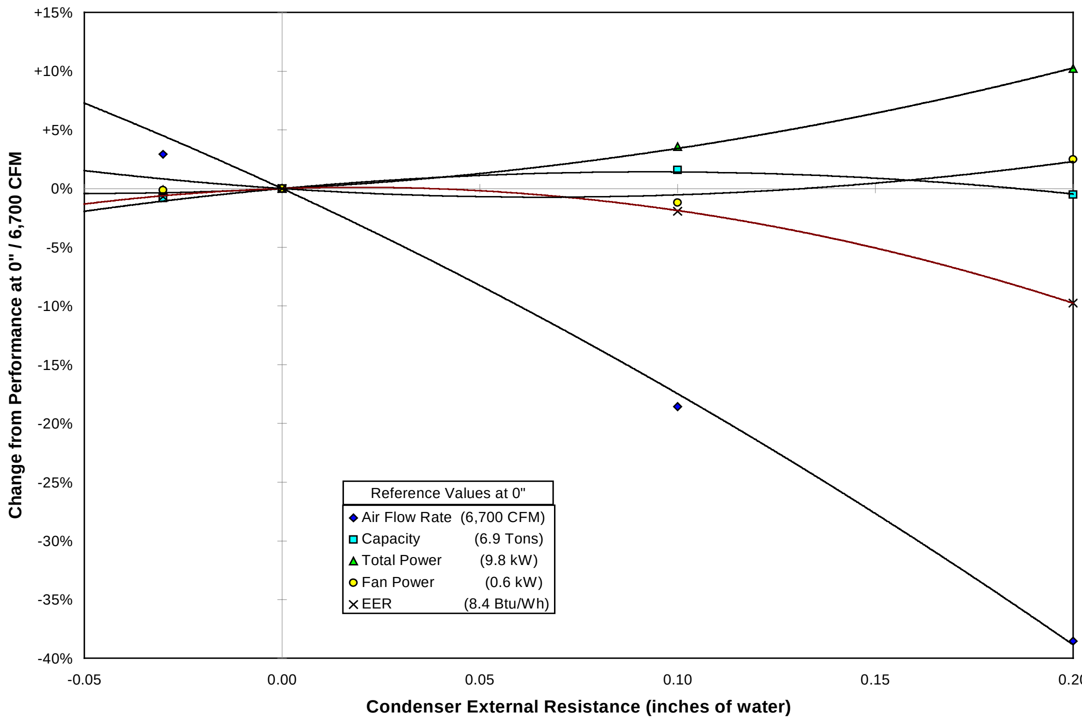

Figure 21. High Efficiency Dual Compressor Unit – Effect of Condenser External Resistance (Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB)

The part-load capability of this system can be examined in a number of ways. The unit can be run with only one compressor to provide part load, or it can be cycled on an off. With the unit running with only one compressor, the steady-state performance can be measured to obtain an integrated part load value (IPLV), which is a measure of performance over a long period similar to a SEER. The test conditions for evaluating the IPLV are slightly different than the ARI Standard Test “B”, with the outside temperature set at 80°F rather than 82°F. The results from the IPLV test with two compressors is similar to the “B” test results, with a capacity of 6.93 tons (versus 7.05 tons average for test “B”), and an EER of 9.44 (versus 9.54). Tests under the same conditions using only one compressor produced an average capacity of only 3.32 tons with a demand of 5.8 kW, resulting in an EER of 6.86. Although the capacity is close to half of the capacity with both compressors running, the lower efficiency of the unit is mainly due to there being no reduction in the fan power as the unit continues to deliver the same airflow. Following the method described in the test procedure, the capacity ratio between running one and two compressors results in a part load factor (PLF) of 0.385, and a resulting IPLV of 7.66. Typically, the IPLV is a bit higher than the EER at the standard rating conditions (in this case 8.40 Btu / Wh). The reason this did not come out that way is probably due to the decrease in capacity at low outside temperatures.

In the tests on the other units, part load conditions were simulated by cycling it on and off under a variety of frequencies. For this unit, the situation is complicated by there being two compressors. Only three sets of cycling performance tests were conducted under different modes of operation, all using the cycling frequency described for ARI Standard test "D". For a direct comparison with the previous units, one set of tests was done where the entire unit was cycled on and off. The results from this test are shown in Figure 22, and in the summary table as tests "C21" and "D21". The results from these tests were used to calculate a degradation coefficient (C_D) of 0.139, and when combined with the results for test "B" give a SEER of 8.87 Btu / Wh.

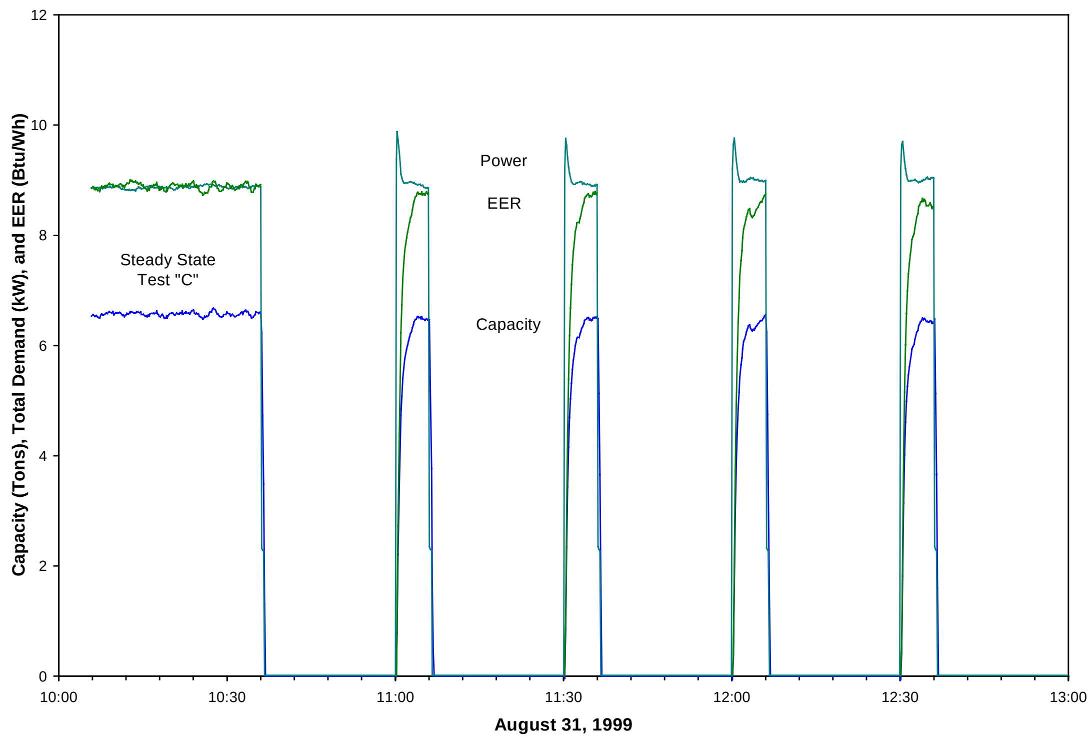

Figure 22. High Efficiency Dual Compressor Unit – Cycling Performance 30 Minute Period, 20% On Fraction (ARI Standard Test "D") (Evaporator Inlet: 80ºF DB, 57ºF WB; Condenser Inlet: 82ºF DB)

However, this mode of operation does not make much sense for a dual compressor unit. In actual use, as the load increases, this unit would start by cycling one compressor until it was on steady, and then cycle the second compressor until it was on steady. Thus two other tests were done, one with a single compressor cycling and the second compressor off ("C1" and "D1"), and another with the first compressor on full and the second compressor cycling ("C2" and "D2"). ARI Standard 210 / 240-94 gives a method for calculating a SEER for a unit having dual compressors using bins of load data weighted to the time that a temperature condition is in effect.

Table 11 gives the bins data from the ARI standard, along with a simulated building load determined from the full capacity at the standard rating conditions (Test “A2”).

Table 11. Bin Data for Determining SEER of High Efficiency Dual Compressor Unit
(from ARI Standard 210/240-94, Table A6.1.2)

Bin No.(j)	Bin Temperature Range (°F)	Representative Temperature for Bin (T_j)	Fraction of Total Temperature Bin Hours (n_j/N)	Building Cooling Load (based on Rated Capacity of 6.91 Tons)
1	65 to 69	67	.214	0.42
2	70 to 74	72	.231	1.46
3	75 to 79	77	.216	2.51
4	80 to 84	82	.161	3.56
5	85 to 89	87	.104	4.60
6	90 to 94	92	.052	5.65
7	95 to 99	97	.018	6.70
8	100 to 104	102	.004	7.74

The capacity and power demand of the unit for each bin temperature are determined by a linear extrapolation of the data from the “A” and “B” tests for each stage of compressor operation. (The analysis uses a linear fit, although the test data does not actually fit one.) If the building load is less than the full capacity with one compressor, the single compressor is cycled on and off. The results from the “C1” and “D1” tests produced a degradation coefficient of 0.111, which is used to calculate an adjustment for the power usage when cycling. When the building load is above the capacity for a single compressor, the second compressor will begin to cycle. The method does not include any penalty for the start up of the second compressor, and simply uses a linear interpolation between the full load values for one compressor and two compressors to determine the capacity and power demand. Once the capacity and demand for each bin have been determined, they are multiplied by the fraction of total hours for that bin and summed. The SEER is then calculated as the ratio of the capacity sum to the demand sum, which for this unit resulted in a value of 7.30 Btu/Wh.

For comparison, the bin method was applied with the degradation coefficient determined from cycling both compressors, and resulted in a SEER of 8.88 Btu/Wh, which compares relatively closely to the simple calculation done without the bin method. As can be seen, the SEER calculation, which takes into account single compressor operation, yields a much lower value. This is due the lower overall system performance during single compressor operation, which in turn is due to the large indoor fan power use whether one or two compressors are operating.

2.5.4 Performance Comparisons

Table 12 provides a summary of test results for all of three units. Comparing the performance of the baseline unit (Test Unit #1) and the high efficiency dual compressor unit (Test Unit #3) is fairly straightforward since they can both be compared as functions of the outside dry bulb temperature. Including the results for the baseline unit with evaporative pre-cooler on the condenser (Test Unit #2) with the others is more difficult because its performance also depends on the outside humidity. It would be possible to include the trends of its performance as a function of the dry bulb temperature of the air leaving the pre-cooler. However, this would make it look considerably worse than the others because of the performance penalties from the airflow restriction and pump power.

Table 12. Summary of Test Results

ARI Test Designation	Parameter	Test Unit # 1 Baseline Unit	Test Unit #2 Baseline Unit with Evaporative Condenser Pre-cooler [5]	Test Unit #3 High Efficiency Dual Compressor Unit
A	Capacity (Btu/hr) (Tons)	88,800 7.40	92,200 7.69	82,900 6.91
Inside: 80°F DB, 67° WB	Power (kW)	11.1	10.7	9.9
Outside: 95°F DB, 75°F WB[1]	EER (Btu/Wh)	8.02	8.58	8.40
B	Capacity (Btu/hr) (Tons)	96,300 8.03	99,600 8.30	84,500 7.05
Inside: 80°F DB, 67° WB	Power (kW)	10.4	10.1	8.9
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	9.28	9.90	9.54
C	Capacity (Btu/hr) (Tons)	87,900 7.32	91,700 7.64	78,900 6.58
Inside: 80°F DB, 57°F WB[2]	Power (kW)	10.2	9.9	8.9
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	8.61	9.28	8.89
D[3]	Capacity (Btu/hr) (Ton-hr/hr)	16,500 1.37	17,500 1.46	14,500 1.21
Inside: 80°F DB, 57°F WB[2]	Power (kWh/hr)	2.2	2.1	1.8
Outside: 82°F DB, 65°F WB[1]	EER (Btu/Wh)	7.49	8.30	7.88
Maximum Operating Conditions	Capacity (Btu/hr) (Tons)	76,000 6.33	87,600 7.30	74,900 6.24
Inside: 80°F DB, 67° WB	Power (kW)	12.1	11.3	11.8
Outside: 115°F DB, 75°F WB[1]	EER (Btu/Wh)	6.26	7.78	6.34
Low Temperature	Capacity (Btu/hr)	86,700 7.23		73,500 6.13

ARI Test Designation	Parameter	Test Unit # 1 Baseline Unit	Test Unit #2 Baseline Unit with Evaporative Condenser Pre-cooler [5]	Test Unit #3 High Efficiency Dual Compressor Unit
Operation	(Tons)			
Inside: 67°F DB, 57°F WB	Power (kW)	9.1		8.0
Outside: 67°F DB, 57°F WB[1]	EER (Btu/Wh)	9.50		9.21
	SEER[4] (Btu/Wh)	8.54	9.26	7.30 (bin)
	IPLV (Btu/Wh)			7.66
Test	Evaporator Airflow (cfm)	3,380	3,320	2,910
Averages	Condenser Airflow (cfm)	5,210	4,770	6,720
	Evaporator Fan Power (kW)	1.83	1.78	2.30
	Condenser Fan Power (kW)	0.64	0.67	0.62
	Pre-Cooler Effectiveness		51%	
Manufacturer's Specifications (At ARI Rating Conditions)	Capacity (Btu/hr) (Tons)	86,000 7.2	93,600 7.8	90,000 7.5
	Power (kW)	9.7	9.5	8.2
	EER (Btu/Wh)	8.9	9.9	11.0
	IPLV (Btu/Wh)			11.6
	Pre-Cooler Effectiveness		60%	

1 Outside wet bulb condition was not maintained for Test Units #1 and #3 since they do not involve evaporation.
2 The low inside wet bulb condition could not always be achieved; coil was likely wet.
3 Test "D" is a cyclic test in which the unit is on for 6 minutes and off for 24 minutes. For the dual compressor unit, the results given are for both compressors cycling together. Single compressor results are given in the main body of the report.
4 Seasonal Energy Efficiency Ratio (SEER) calculated for comparison purposes. It is normally only used as a rating number for units having capacities less than 65,000 Btu / hr.
5 The manufacturer's performance specifications for this system were estimated by combining the evaporative pre-cooler effectiveness specifications and the baseline unit specifications.

Since almost all of the tests on Test Unit #2 were done at the same evaporator inlet conditions, it is possible to compare the test results from the other units at the same conditions to those of Test Unit #2 over a range of humidity. Thus, two sets of comparison figures are provided: one comparing the performance of Test Units #1 and #3 over a variety of evaporator inlet conditions, and one comparing all test units at the same inlet conditions. To improve the clarity of these figures, the curve-fits of the test results have been plotted without the test points on which they are based.

The first set of figures shows the measured cooling capacity. Figure 23 compares Test Units #1 and # 3, while Figure 24 compares all test units. These graphs show the baseline unit exceeded the output of the high-efficiency unit over all of their normal operating range. The curves do come together at the upper limit of the graph at 120°F, but this condition is unlikely to occur. Figure 23 also includes dotted lines showing the least-squares curve fit to the bi-quadratic equation used in the DOE-2 modeling program. This is a way of combining the results into a singe equation, and the result is a set of curves that shadow the fits through the points grouped by wet bulb temperature. The bi-quadratic curves for the 57°F inlet wet bulb are a bit lower than the curves through the data points since the bi-quadratic curve is plotted at 57°F, whereas the other curves use the actual test wet bulb temperatures which were usually higher.

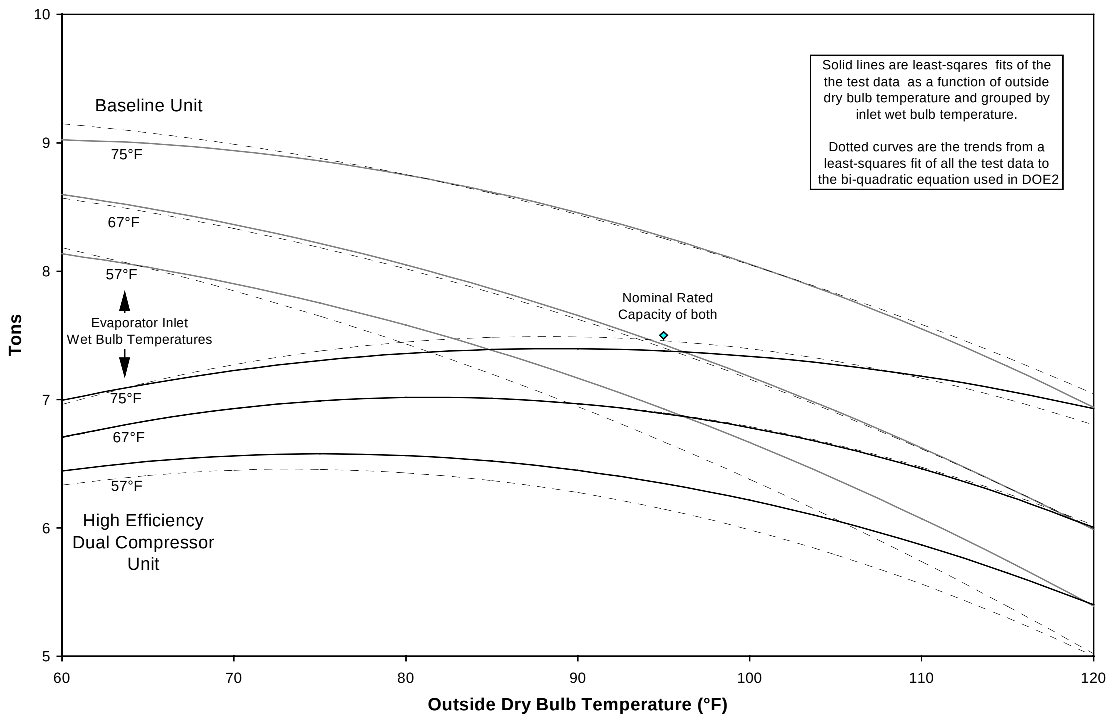

Figure 23. Cooling Capacity Comparison of Baseline Unit and High Efficiency Dual Compressor Unit (Evaporator Inlet: 80ºF DB, 67ºF WB; Condenser Inlet: 95ºF DB)

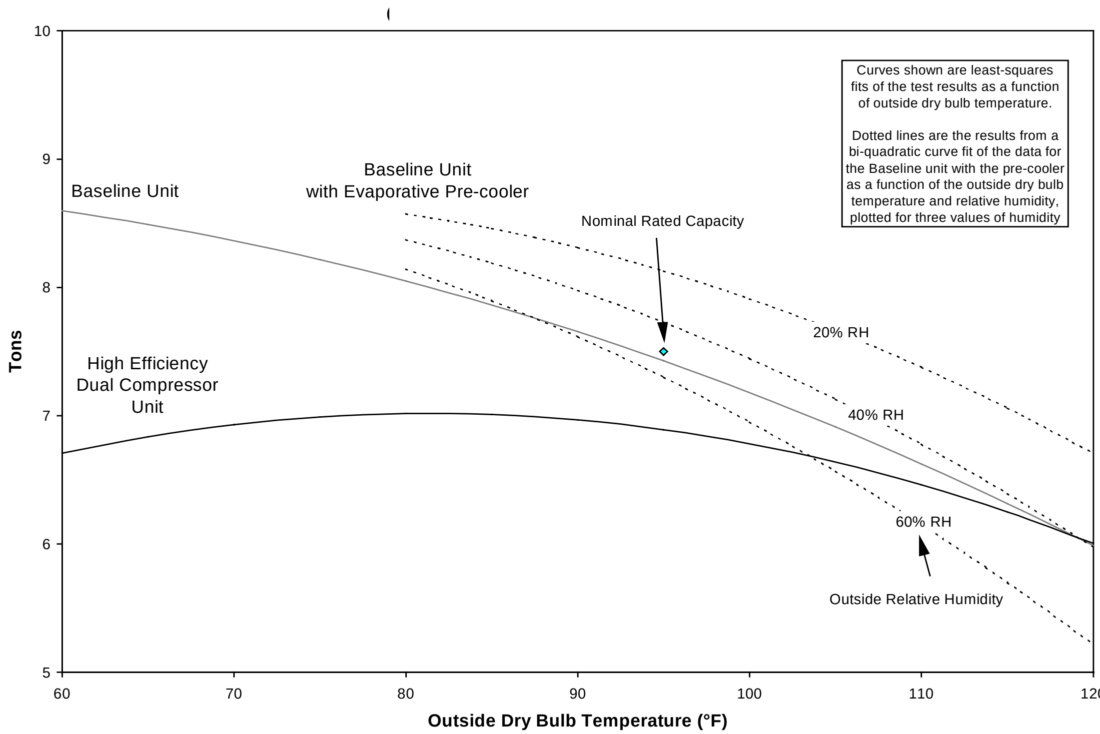

Figure 24. Cooling Capacity Comparison of All Test Units
(Evaporator Inlet Conditions: 80ºF DB, 67ºF WB)

Table 13 lists the coefficients of the bi-quadratic curve fits. (Note: The DOE-2 model normally uses a function that excludes the indoor fan power and its penalty to the capacity. The following equation may be used only if the program is instructed that the fan power is included.)

Table 13. DOE-2 Bi-Quadratic Function Coefficients for Test Units #1 and #3
COOL-CAP-FT = a + b·(ewb) + c·(ewb)² + d·(odb) + e·(odb)² + f·(ewb)·(odb)
ewb = evaporator inlet wet bulb temperature (°F)
odb = condenser inlet dry bulb temperature (°F)

Test Unit	a	b	c	d	e	f
#1	2.276711	-0.033888	0.000252	-0.005402	-0.000051	0.000132
#3	0.501481	0.000463	-0.000037	0.005388	-0.000098	0.000157
Default	0.874030	-0.001142	0.000171	-0.002957	0.000010	-0.000059

The next two sets of figures show the overlays of power demand (Figure 25 and Figure 26) and energy efficiency ratios (Figure 27 and Figure 28). In Figure 25 for power demand, the baseline unit exceeded the usage of the high efficiency unit over the entire test range, although the difference reduced to near zero at the upper temperature limit of the graph. What makes this more significant is that the evaporator fan of Test Unit #3 used 0.5 kW more power than the baseline unit. If Test Unit #3 had an indoor fan with similar performance characteristics as Test Unit #1, the difference in overall unit demand would have been more dramatic.

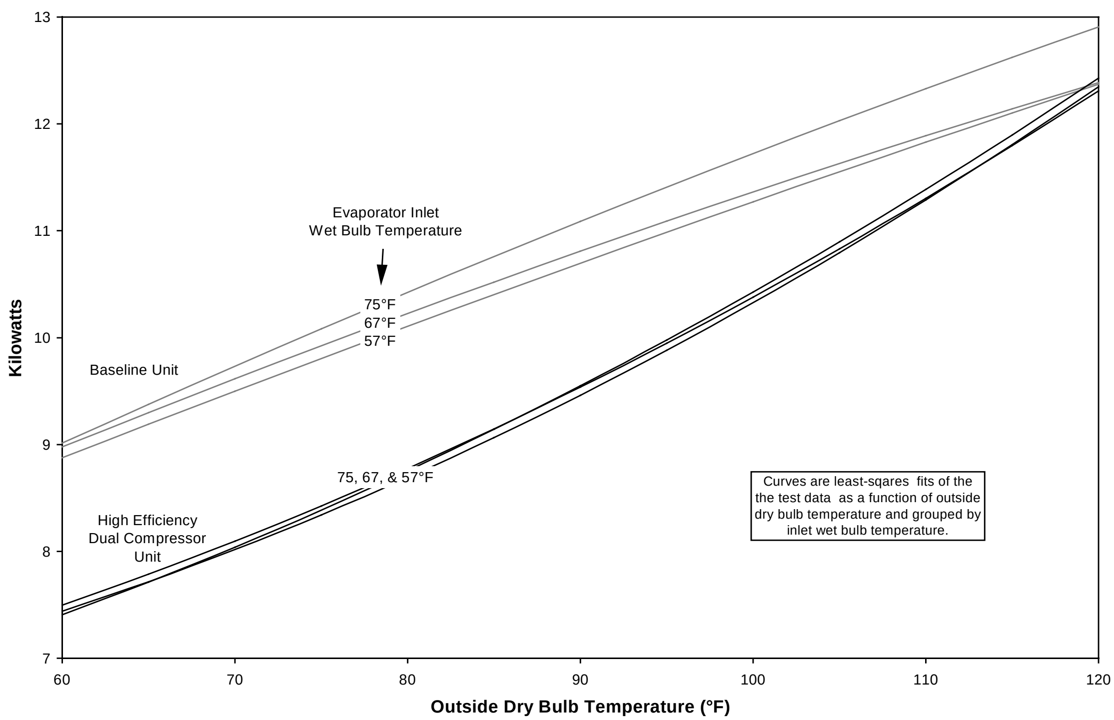

Figure 25. Electric Demand Comparison of Baseline Unit and High Efficiency Dual Compressor Unit (Evaporator Inlet Dry bulb Temperature: 80ºF)

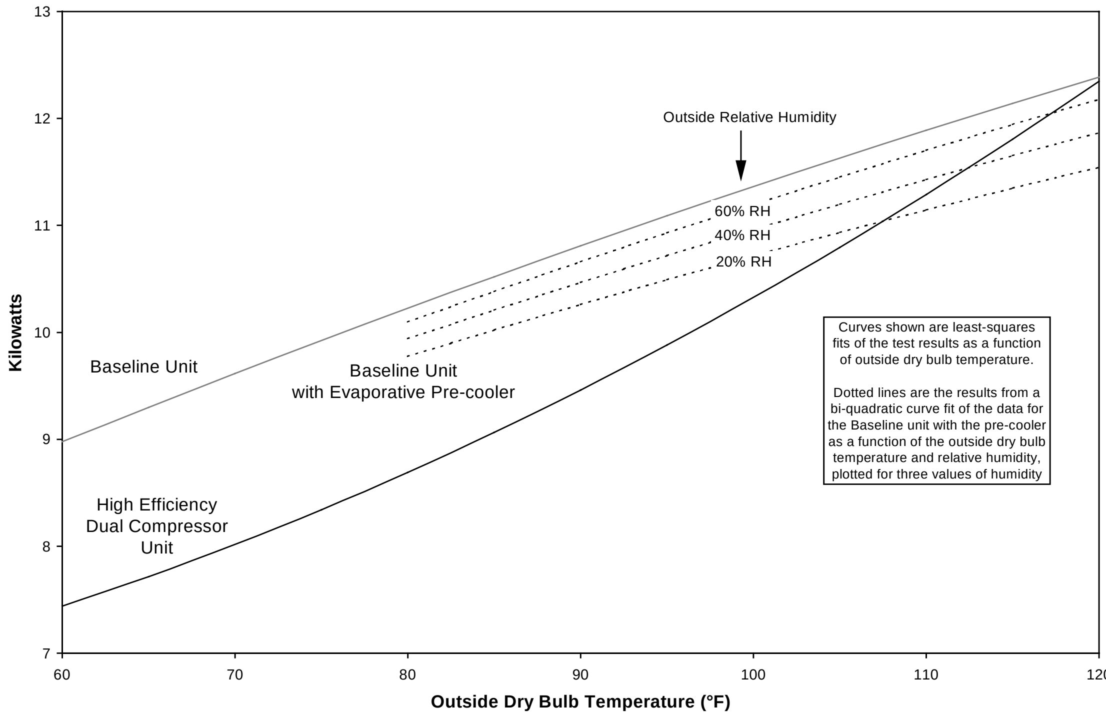

Figure 26. Electric Demand Comparison of All Test Units (Evaporator Inlet Conditions: 80ºF DB, 67ºF WB)

In Figure 27, the lower power draw of Test Unit #3 is offset by the reduced capacity at lower temperatures, which results in energy efficiency ratio trends that are similar for both units. At the design conditions, Test Unit #3 did have slightly better performance.

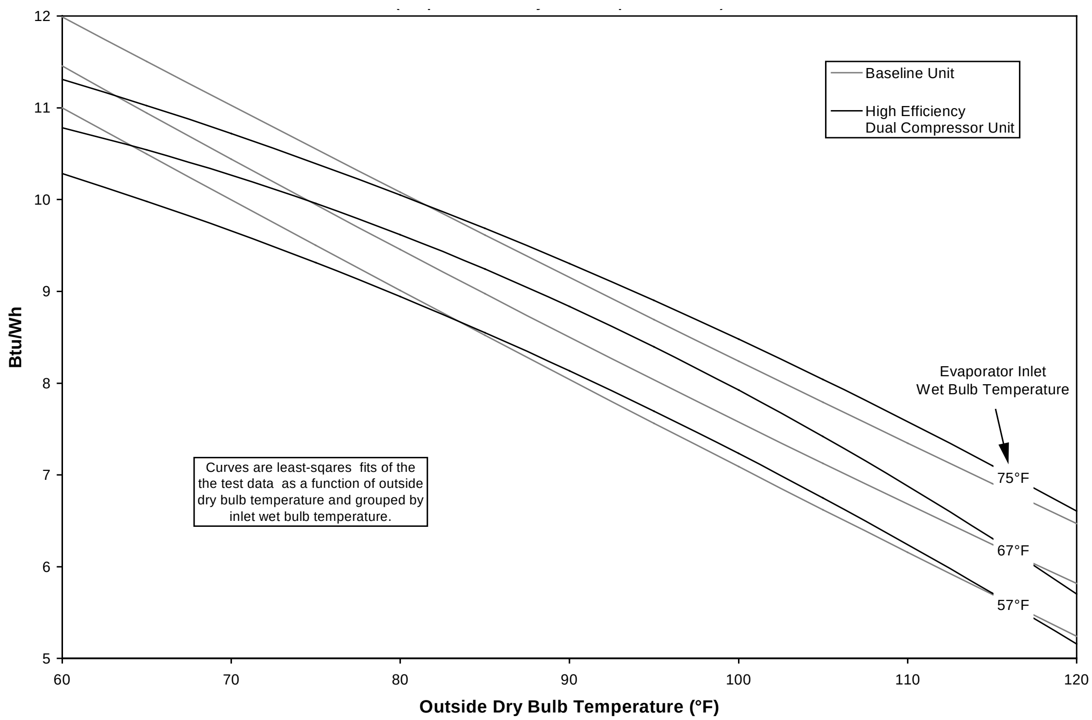

Figure 27. Energy Efficiency Ratio Comparison of Baseline Unit and High Efficiency Dual Compressor Unit (Evaporator Inlet Dry bulb Temperature: 80ºF)

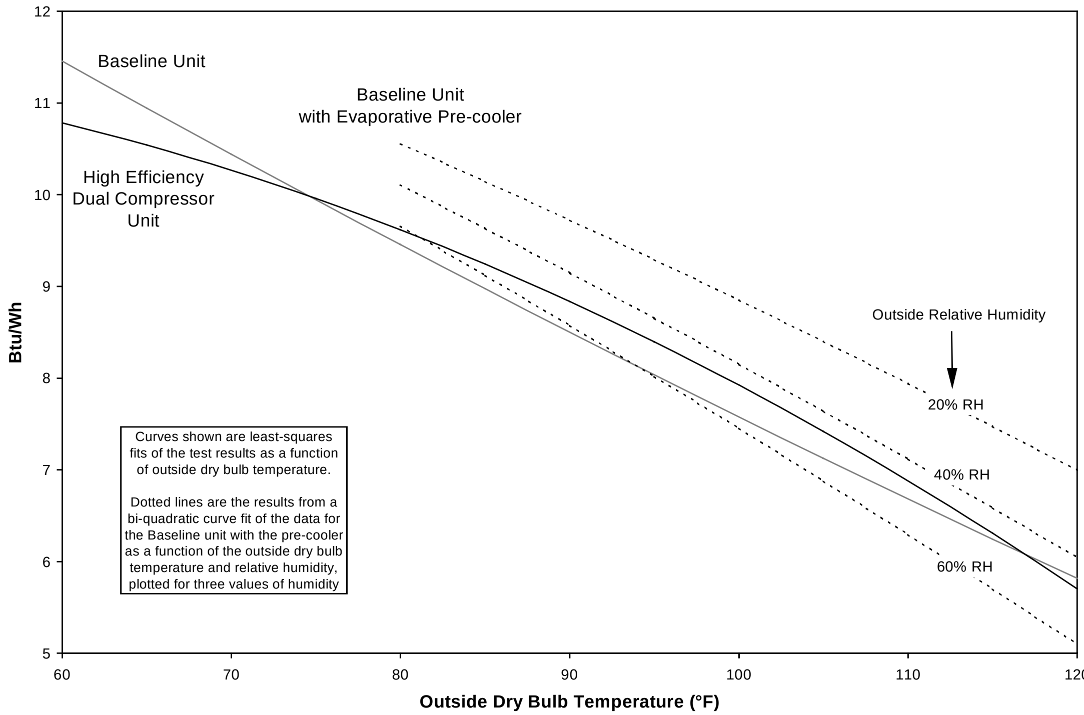

Figure 28. Energy Efficiency Ratio Comparison of All Test Units (Evaporator Inlet Conditions: 80ºF DB, 67ºF WB)

2.6 Uncertainty Analysis

An uncertainty analysis was performed to estimate the uncertainty in the performance factors determined during testing due to measurement errors. The details of that uncertainty analysis are presented in Appendix VI. Below are a summary and discussion of the results of the analysis.

The performance factors determined during these tests were primarily cooling capacity, power use, and energy efficiency ratio. The uncertainty in these performance factors due to measurement system errors is due primarily to uncertainties in the temperature, pressure, humidity, flow, and power measurements. These include bias errors due to instrument calibration and measurement locations (spatial errors), and precision errors due random fluctuations in the measurements (due to either instrument fluctuations or actual process fluctuations).

To perform an uncertainty analysis, the uncertainties in the individual measurements must be estimated. Then, the sensitivity of the final result to those measurements must be determined. Finally, the contributions of each measurement uncertainty must be combined to give an overall uncertainty in the result. Standard statistical techniques for uncertainty analysis were used, such as those described in ASME Power Test Code 19.1-1998 “Test Uncertainty” (ASME 1998), and Ronald Dieck’s “Measurement Uncertainty, Methods and Applications (Dieck, 1992).

The uncertainty analysis resulted in an overall uncertainty in the cooling capacity and EER ranging between about ±3 percent to ±5 percent. At standard test condition “A”, the uncertainty was approximately ±4 percent. At test conditions “B” and “C” it was approximately ±3.5 percent, and at the “max” test condition it approached almost ±5 percent. The uncertainty was reduced during the later tests (after the first test unit) due to improvements in the test system, ranging between ±3 percent to ±4 percent. The uncertainty in the power measurement was about ±one percent.

A heat balance check performed on the unit yielded results that were consistent with the estimated uncertainties. Due to larger uncertainties in the secondary outdoor-side cooling capacity calculations (ranging between ±10 percent to 20 percent), heat balances ranged between 5 percent to 15 percent. The larger uncertainties in the outdoor-side cooling capacity calculation were primarily due to large temperature stratification in the condenser outlet measurement location, some stratification in the condenser inlet, and higher flow measurement uncertainties. Post-test calibration checks also reveled drift in the condenser outlet thermocouple temperature measurements of about 0.6°F, which would account for about four percentage points of the heat balance error.

It should be noted that the uncertainties given above are absolute uncertainties for each unit. The uncertainty in the performance *comparisons* between the units should be much better because many of the uncertainties are common bias’s. As long as no significant drift in instrument accuracy between tests (which there wasn’t for the primary measurements) occurred, and the sensitivities of the performance results to the measurements were not significantly different for the different units (which they weren’t), then the overall bias uncertainty should be similar between units. It is difficult to quantify the relative uncertainty, but it is estimated to be between one percent - two percent based on the repeatability of the results.

2.7 Potential Energy and Demand Savings

An important question regarding the use of various rooftop package unit technologies is the potential pay-back of higher efficiency options. In order to perform this analysis, it is necessary to estimate and compare the energy use of these technologies in real applications. A common method of doing this is to use computer simulation energy analysis programs. While development of a computer model using the results of our testing was not an objective of this project, some estimates were made in order to evaluate the potential for energy savings in California.

A simple building was modeled in the PowerDOE building energy analysis program. PowerDOE is an hourly simulation program that utilizes the DOE-2 simulation engine to perform load and HVAC calculations on buildings in selected locations (Electric Power Research Institute, 1996). The building was an example small office building (approximately 7,000 square feet) with sufficient internal and external loads to require two, 7½ ton packaged air conditioning units was modeled. Therefore, each unit served approximately 3,500 square feet. The building was occupied mainly during the week, with limited occupancy on weekends. This was not a rigorous model, and was not meant to simulate any particular actual building. The purpose was to get an estimate of changes in energy use with changes in the type of packaged unit installed.

Using the results from our tests, the performance curves presented earlier were developed. Using regression analysis on the test results, coefficients for the DOE-2 performance equations were determined. The equations were for the total cooling capacity of the unit, the sensible capacity of the unit, and the Energy Input Ratio (EIR), which is proportional to the inverse of the EER. These factors were described as functions of the indoor wet bulb temperature entering the evaporator, and the outdoor dry bulb temperature entering the condenser.

For the unit tested with the evaporative pre-cooler on the condenser, the model only allows use of a constant effectiveness value for the evaporative cooler. This is then used to calculate the dry bulb temperature entering the condenser, and the same performance relationships for the unit without the pre-cooler are used. For this simple analysis, no attempt was made for more detailed modeling of the evaporative pre-cooler unit. Also, no attempts were made to model the dual compressor operation of the third test unit. The overall EER results for the unit with both compressors operating were used to determine the performance coefficients. An economizer was not included on any of the units.

The building was located in Fresno, California, and the loads on the buildings were modeled to result in a peak load of approximately 12 tons. The model resulted in the unit operating for approximately 2400 hours out of the 8760 hours available in the year. The total building cooling load was approximately 185 million Btu for the year, or 93 million Btu per unit. Although the modeled building utilized two 7½ ton units, the results (Table 14) are given on a per unit basis.

Table 14. Energy Use Comparisons Based on Example Building Model in Fresno, CA

	Test Unit #1 Baseline Unit	Test Unit #2 Baseline Unit with Evaporative Condenser Pre - cooler	Test Unit #3 High Efficiency Dual Compressor Unit
Annual Cooling Energy Use per 7½ ton Unit (kWh/yr)	11,900	10,400	11,400
Cooling Energy Savings (kWh/yr)		1,500	500
Cooling Energy Savings (%)		13%	4%

To evaluate the economics of using higher efficiency options, the cost of electricity and the cost of the options must be incorporated. For this simple analysis, a constant rate of $0.1 / kWh is assumed based on average energy costs. Estimates for the extra cost (equipment and installation) for the higher efficiency options were obtained from a local HVAC contractor.

Table 15 provides a summary of the energy savings compared to the additional costs.

Table 15. Energy Cost Savings and Incremental Cost Estimates for Test Units #2 and #3
Based on Example Building Model in Fresno, CA

	Test Unit #2	Test Unit #3
Energy Cost Savings ($/yr)	$150	$50
Extra Material Cost	$1,317	$1,019
Extra Labor Costs	$1,350	$0
Total Incremental Cost	$2,667	$1,019

In addition to energy savings, there are also demand savings. At the 95°F outside temperature design point, the demand reduction was approximately 0.3 kW for Test Unit #2 and 1.2 kW for Test Unit #3. At peak conditions, the demand reduction is approximately 0.9 kW for Test Unit #2, and approximately 0.3 kW for Test Unit #3. Demand charges vary depending on the rate schedule that a customer is on, so a good estimate of savings for this example is difficult. However, assuming a maximum of about 1 kW, approximately $50/yr additional savings could be achieved based on average demand charges for small commercial buildings.

There are also some potential extra costs associated with the evaporative pre-cooled unit. Water use ranged from about 7 to 14 gallons per hour, depending on the outdoor wet bulb and dry bulb temperature conditions. For the estimated 2,400 hours of operation in the Fresno example, this equates to about 24,000 gallons of water per year (at an average rate of 10 gallons per hour). Any extra costs associated with this, as well as any additional maintenance costs associated with the unit, would have to be considered. The dual compressor unit may some additional benefits, which are hard to quantify. These include better control at low loads with smaller compressors, and possible maintenance cost savings if work or replacement is needed on only one, smaller compressor.

With the savings and costs estimated in these examples, the simple pay-back period for Test Unit #2 in Fresno is about 13 years, and about 10 years for Test Unit #3 (as compared to the Test Unit #1).

Another way of estimating potential energy savings by these units in general without the use of detailed models is to use data for average electric energy use intensities for commercial buildings. The Commercial Building Survey Report (PG&E 1997) gives data of this nature. The overall electric energy use intensity for cooling for all commercial buildings averaged about 3.4 kWh per square foot. However, there was a wide range depending on the type of building. For offices, the value was approximately 3.2 kWh per square foot; for food stores, the value was 5.4 kWh per square foot; and for restaurants, the value was 8.3 kWh per square foot. For the 3500 square foot building served by the 7½ ton unit modeled above, this gives a cooling energy use ranging from about 11,200 kWh (at 3.2 kWh per square foot) to 29,000 kWh (at 8.3 kWh per square foot). The model results for Fresno gave an annual cooling energy use of 11,900 kWh, which is very close to the average value for office buildings.

To estimate the energy savings potential, one needs to know the average EER of the unit over the year. Unfortunately, there is no one performance factor available for this purpose. For smaller air conditioning units, the Seasonal Energy Efficiency Ratio (SEER) is used to give a measure of average performance over the cooling season. The SEER is typically not calculated for units above 5.4 tons (65,000 Btu / hr). However, with the tests performed on these units, the SEER can be calculated. This was done for each unit in the results section discussed above. Comparing the SEER value of each unit against the Baseline Unit, one can calculate a percentage increase. Applying this percentage increase to the annual cooling energy use for the Baseline Unit, estimated annual energy savings could be calculated (Table 16).

Table 16. Estimate of Annual Energy Savings Based on Comparison of SEER Values
(based on a Baseline annual energy use of 11,900 kWh)

Test Unit	SEER (Btu/Wh)	Increase over Baseline (%)	Estimated Energy Savings (kWh/yr)
#1 (Baseline)	8.54		
#2 (Evap Pre-cooler)	9.26	8.4%	1,000
#3 (High Efficiency), with individual compressor control and cycling	7.30 (bin)	-14.5%	-1,728
#3 (High Efficiency), with both compressors cycled together	8.87	3.9%	460

The energy savings estimate for the evaporative condenser pre-cooled unit based on SEER comparisons is about 35 percent lower than estimated by the model. It does make sense that the SEER value would not necessarily be a good indicator of annual performance for the unit with an evaporative pre-cooler. This is because the performance of these systems is highly dependent on the weather conditions (specifically, the wet bulb depression). The standard SEER calculation does not take into account variations in the outdoor humidity, so it would not characterize the benefits of operation in hot, dry climates.

For the high efficiency dual compressor unit, the SEER value using the “bin” method is lower than the baseline unit, which predicts an annual energy increase. The SEER calculation for dual compressor units using the bin method makes use of an assumed reference outdoor temperature distribution, an assumed building load profile, and uses single or dual compressor operation and cycle performance values depending on that load profile. This is outlined in ARI Standard 210 / 240, as well as ASHRAE Standard 116. This results in a significant percentage of the time operating with a single compressor. And because the system performance with a single compressor operating was relatively low, the SEER value comes out low. This also agrees fairly well with the IPLV result (7.66), which is another measure of part load performance. There is no comparable value for the single compressor units, but comparing it to the SEER value for the baseline unit again shows a lower performance for the high efficiency dual compressor unit.

If the dual compressor unit is allowed to cycle both compressors at the same time, the SEER value comes out higher (8.87), and shows about a four percent improvement over the baseline unit. This agrees well with the model results for the example building in Fresno.

These SEER comparison results for the high efficiency dual compressor unit leads to the conclusion that the energy savings potential is very dependent on both the full load and part load performance, and method of control of these units. If single compressor operation has a relatively low EER, it is better to operate both compressors together and cycle them on and off. This conclusion does not address long-term effects of compressor cycling.

The energy use intensity of the small office building discussed in this example was at about the average for all commercial buildings (3.4 kWh / square foot). Many small commercial buildings that would use these technologies have larger energy use intensities. As discussed above, restaurants average about 8.3 kWh / square foot. At this rate, an equivalent size building would have an annual cooling energy use of about 29,000 kWh / year. Assuming the Test Unit #2 technology could save the same 13 percent as the small office model, the annual savings would be about 3,800 kWh / year. This results in a pay-back period of about 6 years if one unit could meet this load. For Test Unit #3, an equivalent four percent savings would be 1,200 kWh / year, also resulting in a pay-back period of about six years.

An obvious conclusion from the discussion above is that the benefits of using higher efficiency options depend on the type of building, where it is located, the cooling load, and the cost of the higher efficiency option. Applications need to be looked at in specific locations in order to determine the value of performance improvements. There is also the issue of new installations compared to retrofits. In installing a new unit (as a replacement or initial installation), it may be more cost effective to go with a high efficiency unit. However, an evaporative pre-cooler can be added to an existing unit, increasing the performance without having to purchase a new unit.

The model described above was a simple model aimed at producing some rough comparisons between options. It is also obvious from the above discussion that more work needs to be done to better characterize the annual performance of these units in various locations, either by improved models, or more representative performance factors analogous to the SEER rating.

Based on a California Energy Commission Staff Report on California Energy Demand (California Energy Commission 1995), annual electric energy use for cooling of commercial buildings in PG&E's service territory alone in 1998 is estimated to be approximately 3,500 GWh / yr. The Commercial Building Survey Report estimates that packaged electric cooling equipment accounts for approximately 66 percent of the installed 3.53 million tons of cooling capacity used by PG&E's commercial customers. This equates to approximately 2,600 GWh / yr of electric use for cooling of commercial buildings with packaged units. It is also estimated that approximately 66 percent of this commercial building energy use is for small commercial buildings (offices, restaurants, retail stores, food stores), with an energy use of approximately 1,700 GWh / yr. Electric energy use for Southern California Edison (SCE) and San Diego Gas and Electric (SDG&E) add up to numbers similar to PG&E's (based on utility electric load data). Therefore, the total electric energy use for packaged air conditioning units on small commercial buildings in California is on the order of 3,400 GWh / yr (for the three large utilities' service territories). The percentage of this energy use that could be saved by incorporating these technologies depends on the number of replacements to be expected, and the number and location of units that could add on evaporative pre-cooler on the condensers.

3.0 Conclusions and Recommendations

3.1 Performance Test Results

The test units were "off-the-shelf" production units that would typically be selected for a commercial application. This means they were not subject to any special construction by the manufacturer or features that would not normally be installed in the field. It also means they are subject to the normal variations in manufacturing that can make units of identical design perform differently. Thus, the performance of these particular samples may not represent the average performance of these models.

Baseline Unit (Test Unit #1). The baseline unit achieved its rated capacity but did not meet the rated EER at the nominal ARI rating conditions. This might be because of additional power used by the indoor fan. Because it was a high heat model, the unit tested came with a standard 2 hp motor. However, even with this fan motor, the manufacturer did list the ratings given in Table 2 in the previous section. We were unable to get verification from the manufacturer, but must assume that the unit was rated with a lower RPM, lower HP fan in order to achieve the rated EER. Our tests were run at higher indoor air flow rates than the ARI rating condition (3,300 cfm versus 2,625 cfm). However, sensitivity tests run at the lower air flow rate by increasing the external static pressure showed little change in the EER.

Baseline Unit with Evaporative Pre-cooler (Test Unit #2). Under most typical outdoor conditions, the addition of an evaporative pre-cooler produced increased capacity and reduced power demand, resulting in an increased efficiency. The benefit achieved from the pre-cooler is highly dependent on the outside relative humidity. Comparing the test results at the ARI Standard rating conditions, where the baseline unit achieved a capacity of 7.40 tons, the pre-cooler increased the capacity to 7.69 tons at 40 percent relative humidity - an increase of four percent. At these same conditions, power demand was reduced by four percent (from 11.1 to 10.7 kW), with the resulting efficiency improved by 7 percent (from 8.02 to 8.58 Btu / Wh). At the hot, dry conditions ("Maximum Operating Conditions, 115°F dry bulb, 15 percent relative humidity), the capacity is increased by 15 percent, power is reduced by 7 percent, and the efficiency increased by 24 percent. However, there is a level of humidity above which there is actually a reduction in performance due to the penalties caused by increased airflow restriction and pump power. This upper humidity limit is dependent on the outside temperature: at 82°F, the relative humidity can get as high as 62 percent before there is no improvement, but at 115°F, the humidity must be less than 42 percent.

High Efficiency Dual Compressor Unit (Test Unit #3). For this unit, the test results showed a lower capacity at the rating conditions than its published value of 7.5 tons, and a lower capacity in comparison with the baseline unit (6.91 tons versus 7.40). But because of the high efficiency compressors, the power demand was significantly reduced compared with the baseline unit (9.9 kW versus 11.1), resulting in a slightly better efficiency (8.4 Btu / Wh versus 8.0). However, the EER of the unit was still significantly below its rated EER at the nominal ARI rating conditions. Higher than expected indoor fan power again may have been the major reason for the efficiency shortfall. Similar to the baseline unit, this unit came with a standard 2 hp motor. Once again, we were unable to get verification from the manufacturer, but must assume that the 11.0 rated EER was achieved with a lower RPM, lower HP fan. The capacity of the unit also dropped off at lower outside temperatures, possibly due to an overly restrictive expansion orifice. The performance of this unit may be improved by using a variable thermal expansion valve and by use of a variable speed fan.

Although the primary objective of this project was to compare the performance of various technology options to each other, some interesting observations were made when comparing the measured performance of the units to their ARI rated values. One lesson that was learned through the testing is that manufacturers may have their products rated with a lower airflow rate and lower rpm indoor fan than what the unit will normally require in the field. This meets the ARI requirements as long as it is run against the minimum external static pressure specified in the standard (0.25 inch of water for this size unit; from Table 6 in ARI Standard 210 / 240-94). Since fan power increases as the cube of the airflow rate and speed, rating its performance with a smaller fan can result in both a greater capacity and lower total power demand, creating a higher efficiency rating. Unfortunately, it is not always obvious from specification sheets which fan motor and drive was actually used during rating tests. Therefore, when comparing the performance of these units, it is important to know which fan motor and drive will be supplied and its effect on overall system performance, so adjustments to the specified ratings can be made for the actual indoor fan to be used.

3.2 Energy and Demand Savings

By improving the efficiency of rooftop packaged air conditioning units, there exists significant potential for electric energy and demand savings in California. Section 2.7 in the Discussion Section of this report gives details on some estimates that were made concerning the savings potential from using the technologies evaluated in this project. Here is a summary of those results.

The energy and demand savings from using the evaporative pre-cooler on the condenser technology (Test Unit #2) are very dependent on the outdoor dry bulb and wet bulb temperature conditions, and therefore on the location. Using a very simple model, a 7½ ton unit operating on an example office building in Fresno, California, was estimated to result in energy savings of about 1,500 kWh / year, or about 13 percent of the cooling energy used by a baseline unit without the evaporative pre-cooler. The high efficiency dual compressor unit was estimated to result in energy savings of about 500 kWh / year, or about four percent of the baseline energy use. However, this was based on a simple model with both compressors controlled together. Energy savings with individual compressor control may actually be less due to lower system EERs with one compressor operating. On the other hand, buildings with higher cooling load intensities, such as restaurants, could experience significantly higher energy savings than those estimated above.

Compared to a baseline unit, added equipment and installation costs for evaporative pre-cooler on the condenser were estimated to be about $2,700 for a typical installation. Added equipment costs for the high efficiency dual compressor unit (no extra installation costs) were estimated to be about $1,000.

Demand savings also depended on weather conditions. At 95°F dry bulb temperature, and 75°F wet bulb temperature (standard ARI rating conditions), demand savings were about 0.4 kW for the evaporative pre-cooler on the condenser unit, and 1.2 kW for the high efficiency dual compressor unit. However at extremely hot and dry temperature conditions (115°F dry bulb and 75°F wet bulb), demand savings were 0.8 kW for the evaporative pre-cooler on the condenser unit, and 0.3 kW for the high efficiency dual compressor unit. One other advantage of the dual compressor system is reduced demand when only one compressor is required (lowering demand by approximately 3 kW). However, both compressors are likely to be required during peak weather conditions. Once again, the appropriate technology for an application depends on conditions at the site (load, weather, etc.).

Annual energy use comparisons were made in two ways. The first used an hourly simulation computer model incorporating the laboratory performance test results on a simple example office building. The second used a comparison of the calculated Seasonal Energy Efficiency Ratio (SEER) based on the laboratory test results. For the evaporative pre-cooler on the condenser unit, the SEER comparison of energy savings was about 35 percent lower than the model results, giving an 8.4 percent reduction instead of 13 percent. This is because this unit's performance is very dependent on the temperature and humidity at the site, and this dependence is not captured by the SEER calculation.

For the high efficiency unit, the SEER comparison predicts an average energy *increase* over the baseline unit. The difference is most likely due to the model not appropriately characterizing the dual compressor operation. The system EER during single compressor operation is lower than for dual compressor operation because of the large auxiliary energy use (i.e., the indoor fan). The BIN method used for the SEER calculation results in a significant proportion of the time being run in single compressor mode, thereby lowering the average EER for this unit. In addition, site-specific weather conditions for the model are different than those used in the standard SEER calculation. It appears that the energy savings potential of this particular unit depends on how it is controlled for dual compressor operation. More work needs to be done to accurately estimate potential energy savings for this unit in actual installations.

3.3 Benefits to California

Overall estimates of potential energy savings for California were also made. Based on survey data for California, approximately 3,400 GWh per year of electric energy is used by packaged air conditioning units on small commercial buildings. Based on an average system energy rate of $0.1 per kWh, this represents $340 million per year in electric energy costs. The percentage of this number that can realistically be saved by these technologies depends on the number of replacement units to be installed, and the locations that could be retrofitted with evaporative pre-cooler on the condensers.

The use of an evaporative pre-cooler on the condenser inlet air, and use of high efficiency dual compressor technologies provide both energy and demand savings for California's small commercial customers. If an average of five percent improvement in efficiency was achieved across all installations, then there is the potential for 170 GWh / yr of savings in California (based on an estimated 3400 GWh / yr of electric energy used by packaged air conditioners on small commercial buildings). The potential demand reduction can be on the order of 100 MW if even the minimum two percent demand reduction under very hot conditions is applied to the estimated three million tons of installed packaged unit cooling capacity on small commercial buildings in California.

While these savings are potentially significant, the savings achievable through the market adoption of more advanced technologies and better design practices are much larger. Improved design practices can decrease fan power by half and thereby reduce individual rooftop package unit demand by 1 kW. Because fans operate year around, the kWh savings achieved by a 1 kW reduction in fan power will be larger than the five percent savings from the two technologies evaluated in this project. The benefit of this research to California will be captured when equipment manufacturers produce equipment specifically designed to operate efficiently in hot dry climates and engineers design low static air distribution systems.

3.4 Recommendations

One truth about performance testing in a research environment is that it often creates as many or more questions than it answers. These tests were no exception. This project identified several areas where further work would be valuable in improving the energy efficiency of rooftop package unit applications in California. These are summarized below.

- Work needs to proceed to address the issue of indoor-side fan energy use. As discussed in this report, these components use a relatively large percentage of the total unit energy (up to 30 percent), and are very dependent on the installation conditions. Improvements in these component efficiencies would help improve overall unit EER. Further study areas would be to examine the cost effectiveness of higher efficiency motors that produce less heat, relocating the fan motors out of the conditioned air stream, and two-speed or variable speed fans that can adjust to the load.
- Improved clarity is needed for the consumer in using and comparing ARI performance ratings. As indicated above, the indoor-side fan energy can significantly influence the EER rating for a unit, and it is not always clear which components were included in the rating test. Consumers may not always get what they think they're getting in terms of overall EER unless they are careful about some of the specifications, and their actual installation conditions. For example, substituting a 2 hp fan for a 1 hp fan can drop the EER of a unit from about 9 to 8 (10 percent reduction) or lower, depending on the flow and static pressure conditions.
- Comparisons of energy and demand savings of these and other technologies could be improved with further efforts in computer modeling using simulation programs. These could help produce guidelines for the consumer for comparing different technology options in different locations in California. In addition, better annualized performance factors, analogous to the SEER rating, could be developed to provide an easier way to compare technologies.
- Additional high efficiency technologies could be evaluated (through testing and modeling) to improve the knowledge base on real performance of these systems over a wide range of operating conditions. Examples of other technologies were discussed in Appendix I, and include such things as economizers; pre-cooling supply air using direct and/or indirect evaporative cooling, heat wheels, or heat pipes; improved fan efficiency; or other component efficiency upgrades (heat exchangers, compressors, expansion devices, etc.).
- A number of tests were conducted which looked at the changes in performance as affected by the external resistance on both the evaporator and the condenser. Fouling of

the heat exchanger coils can add to this resistance and reduce the system efficiency. While filters help keep the coils of the evaporator clean, the condenser coils are exposed to the elements and can be clogged by anything carried on the wind. Testing can examine what effect this fouling has on a system, and what improvement can be gained through coil cleaning. Also, what cleaning method is the most effective, and what the recommended frequency of cleaning should be, could also be examined.

3.5 Market Transformation

The roof top air conditioning unit (RTU) market is very competitive and, above all, cost driven. National manufacturers want to produce the least number of models necessary to meet the demands of a single national market. The market cares little about the quality or efficiency of the equipment; it is primarily concerned that the equipment provides cold air. Customers in most cases do not have the ability to determine if they are receiving value for their purchase and operating dollars.

Transforming the RTU Market will require a multi-phased effort that includes:

- Building owners, design engineers, facility engineers and architects will have to be educated about how RTUs perform and the effects building components such as ducts and roof top temperatures have on RTUs.
- New technology demonstrations will have to be conducted and case studies presented to show building owners, design engineers, facility engineers and architects and contractors that efficient cost effective options exist.
- Market Transformation Programs will have to be established that address the market barriers which impede the widespread adoption of the more efficient technologies.
- Codes and standards will have to be revised to reward and support the use of high efficiency technologies.

The information developed through this project will be used in addressing the first steps in RTU market transformation by educating building owners, design engineers, facility engineers and architects. Additionally, initial steps toward the creation of Market Transformation Programs designed to reduce the market barriers facing high efficiency RTUs will be taken.

Specific activities include:

- Project results will be incorporated into the Package Units: Selection, Specification and Control for Optimal Performance course at Pacific Gas and Electric Company’s Pacific Energy Center (PEC) on December 1, 1999.
- During the year 2000 the Package Unit course will be presented an additional time at the Pacific Energy Center. Plans are being made for additional presentations of the course at other location(s) outside San Francisco.
- The project Final Report will available to the public through the California Energy Commission’s PIER and the PEC websites.
- Project results will be presented to PG&E Commercial RTU Market Transformation Program managers to determine the potential of establishing field demonstrations for promising new technologies.

4.0 Glossary

ARI	Air-conditioning and Refrigeration Institute.
ASHRAE	American Society of Heating, Refrigeration and Air-conditioning Engineers, Inc.
Capacity	The rate, expressed in Btu / h or Tons, at which the equipment removes heat from the air passing through it under specified conditions of operation.
Coil bypass factor	A measure of the fraction of airflow that is not cooled by the evaporator coil. If the outlet air is considered a mixture of the entering air and saturated air at the coil temperature, then this is the ratio of the difference between the outlet dry bulb temperature and the coil temperature to the difference between the inlet dry bulb temperature and the coil temperature. The coil temperature is found by extrapolating a line from the entering to the leaving air conditions plotted on a psychometric chart until it intersects with the saturation line. If the line does not intersect the saturation line, or if there is sensible cooling only, the coil bypass factor is undefined.
Coil, indoor	The heat exchanger which removes heat from the conditioned space (evaporator).
Coil, outdoor	The heat exchanger which rejects heat to a source external to the conditioned space (condenser).
Cooling load factor (CLF)	The ratio of the total cooling done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady-state cooling done over the same period at constant ambient conditions.
Cyclic test	A test where the indoor and outdoor conditions are held constant, but the unit is manually turned "on" and "off" for specific time periods to simulate part-load operation.
Degradation coefficient (C_D)	The measure of the efficiency loss due to the cycling of the unit.
Demand	Electrical power input, expressed in kW.

Dry-coil test	A test conducted at an intake wet bulb temperature and a dry bulb temperature such that moisture will not condense on the evaporator coil of the unit (i.e. The dew-point temperature of the inlet air is less than the evaporator coil temperature).
Energy efficiency ratio (EER)	A ratio calculated by dividing the cooling capacity in Btu / h by the power input in watts at any given set of rating conditions, expressed in Btu / Wh.
Integrated part load value (IPLV)	A single representative number for efficiency at partial loading.
Latent cooling	The amount of cooling in Btu's necessary to remove water vapor from the air passing over the indoor coil by condensation during a period of time. (Energy equivalent to the reduction of moisture without a change in temperature.)
Part load factor (PLF)	The ratio of the cyclic energy efficiency ratio to the steady-state energy efficiency ratio at identical ambient conditions.
Performance	All-encompassing term referring to values of capacity and EER.
Pre-cooler effectiveness	The dry bulb temperature reduction achieved by the air passing through an evaporative pre-cooler divided by the entering wet bulb depression.
Published rating	A statement of the assigned values of those performance characteristics, under stated rating conditions, by which a unit may be chosen to fit its application. These values apply to all units of like nominal size and type (identification) produced by the same manufacturer. The term "published rating" includes the rating of all performance characteristics shown on the air-conditioner or published in specifications, advertising or other literature controlled by the manufacturer at stated rating conditions.
Rating conditions	Any set of operating conditions under which a single level of performance results, and which causes only that level of performance to occur.
RTU	Roof top unit air-conditioner

Seasonal energy efficiency ratio (SEER)	The total cooling of a central air-conditioner in Btu's during its normal usage period for cooling divided by the total electric energy input in watt-hours during the same period. Normally only determined for units with capacities less than 65,000 Btu / hr.
Sensible cooling	The amount of cooling in Btu's performed by a unit over a period of time, excluding latent cooling. (Energy equivalent to the reduction of air temperature without a change in moisture.)
Single package unit	Any central air-conditioner in which all the major assemblies are enclosed in one cabinet.
Steady-state test	A test in which all indoor and outdoor conditions are held constant within prescribed tolerances and the unit is in a non-changing operating mode.
Temperature, dry bulb	The temperature of the air as indicated by an accurate thermometer corrected for radiation effects.
Temperature, wet bulb	The temperature an air sample would reach if brought to saturation adiabatically through the evaporation of water. It is the temperature indicated by a thermometer with its sensing element wrapped in a moistened sheath and exposed to a minimum air velocity. (Used in combination with a dry bulb thermometer to form a "psychrometer" for the measurement of humidity.
Ton	A term of cooling capacity equivalent to 12,000 Btu / hr.
Wet bulb depression	The difference between the dry- and wet bulb temperature of an air sample.
Wet-coil test	A test conducted at an intake wet bulb and a dry bulb temperature such that moisture will condense on the test unit evaporator coil (i.e. the entering dew-point temperature is above the evaporator temperature).

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Appendix I

Task 3 Interim Report – Research of Available Technologies

Appendix II

Task 4 Interim Report – Test and Evaluation Plan

Appendix III

Measurement System Descriptions

Appendix IV

Test Data

Appendix V

Detailed Test Unit Specifications

Appendix VI

Uncertainty Analysis

APPENDIX I

Task 3 Interim Report
Research Available Technologies

Evaluation of Small Air Conditioning Units for Northern/Central California Climates

Task 3 -Research Available Technologies

CEC Contract # 500-97-10

Project #1

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Packaged Rooftop Air Conditioner Technologies

Executive Summary

Packaged rooftop air conditioners are installed on thousands of small commercial buildings throughout California. They are a relatively inexpensive means of providing space cooling and heating for these buildings. They are designed to be rugged and require only occasional maintenance, but they typically have low energy efficiency. This situation, however, has been mitigated by minimum efficiency standards imposed by State and Federal governments. These standards now set a lower limit on the energy efficiency of all air conditioners sold in California.

Higher efficiency rooftop air conditioners, however, are available on the market. Available data show that they have significantly higher efficiencies than that required by minimum efficiency standards. In addition, there is equipment available that can be added to any rooftop air conditioner to likewise improve its efficiency.

The overall scope of this project is to identify those technologies which can potentially improve the efficiency of packaged rooftop air conditioners, and document the actual energy performance of some selected technologies through laboratory testing.

This report summarizes the first part of this work, and is the deliverable for Task 3. Specifically, this task is to identify and assess technologies which have the potential to improve the energy efficiency of the packaged rooftop air conditioners in Northern and Central California climates. This assessment would result in the selection of one to three technologies for testing and evaluation.

This investigation highlighted the following technologies for consideration:

- Introducing cool outside air into the air conditioner with an economizer to minimize the enthalpy of the air entering the evaporator, thus reducing compressor operation.

- Pre-cooling outside air entering the air conditioner using an indirect or combination of indirect and direct evaporative cooling equipment, heat wheel, or heat pipes.

- Combining the high efficiency compressors with increased evaporator and condenser surface areas and multiple compressors in a high efficiency air conditioner.

- Lowering the condensing temperature at high outdoor air temperatures using an evaporative pre-cooler.

- Increasing the evaporator capacity while potentially decreasing electrical input to the compressor by incorporating refrigerant sub-cooling into the air conditioner.

Based on the information obtained in this investigation, and on discussions with HVAC experts within PG&E, it was decided to test a high efficiency dual compressor air conditioner with and without an evaporative condenser pre-cooler. A baseline 7.5 ton air conditioner was chosen for testing to represent the performance of packaged rooftop air conditioners which meet minimum energy efficiency requirements

This research effort attempted to identify the most common capacity air conditioner installed on small commercial buildings. Opinions among experts, and national data for air conditioner shipments were considered. Although the information obtained did not result in a clear-cut consensus on the most common size, it was decided to test 7.5 ton capacity air conditioners. There were several reasons for this decision, including the increased impact of 7.5 ton units on energy consumption and demand as compared to 5 ton units, the availability of both single and dual compressor configurations and other high efficiency options.

Packaged Rooftop Air Conditioner Technologies

INTRODUCTION

Currently in California, a new air conditioner purchased for installation on a commercial building must be certified by the manufacturer to meet minimum energy efficiency standards. To satisfy these standards, manufacturers have incorporated certain technologies into these air conditioners. Other technologies exist, however, which could significantly improve upon the currently required performance and are believed to be potentially viable in the market. PG&E and the California Energy Commission are interested in evaluating the performance of such technologies both in terms of their energy consumption and their impact on electrical demand. The scope of this project is to identify these technologies and to evaluate the energy performance of selected technologies under laboratory type conditions which simulate operating conditions in PG&E's service territory.

The focus of this project is on technologies that will improve on the performance of packaged rooftop air conditioners installed on small commercial buildings. These buildings include small retail stores such as those found in a strip mall, restaurants, grocery stores, convenience stores, small office buildings, theaters, and some public buildings such as school buildings. Packaged rooftop air conditioners incorporate a compressor, an evaporator coil, evaporator fan (also known as an indoor fan) a condenser coil, a condenser fan (also know as an outdoor fan),and controls contained in one metal box. Single and multiple small capacity rooftop air conditioners are typically used to cool and heat these small commercial buildings. They are relatively simple and inexpensive to install, to operate and maintain.

The first phase of this project was an effort to determine the capacity of air conditioners most commonly installed on small commercial buildings. Different sources provide conflicting information, and unfortunately, shipment data is not segmented by state. Based on this research, it was decide to investigate air conditioners having an approximate nominal cooling capacity of 7.5 tons. A single 7.5 ton unit is small enough to cool a relatively small commercial building, while multiple such units might be used on larger small commercial buildings. The 7.5 ton air conditioner is the smallest capacity unit which can be purchased with multiple compressors. This is important in matching the unit's capacity to the air conditioning load at part load conditions.

This project considers air conditioners which would be installed on new or existing buildings in the PG&E service territory. While air conditioners are designed to operate in climates which are hot and humid, weather conditions in the PG&E service territory are hot and relatively dry. Therefore, there are opportunities to apply technologies which work well in such a climate.

A survey of the industry was made to find equipment which incorporates such technologies. A list of technologies and the equipment which incorporates each is shown in Table 1. The impact of each technology on electricity consumption and demand depends on specifics of the building on which an air conditioner is installed. The relative impact of some technologies, however, is indicated by the energy efficiency ratio (EER) and integrated part load value (IPLV) for the equipment shown. Both are indicators of an air conditioner's energy efficiency, in Btu/h per watt, at specific operating conditions. EER is evaluated under near design operating conditions; while IPLV is an indicator of a unit's efficiency at part load operating conditions.

Electricity consumption savings and demand reduction potential are only two criteria that may be used to determine what equipment to test. Other attributes that are listed are physical size, cost and availability of equipment along with the potential applicability of the particular piece of equipment.

Some technologies listed supplant a portion of the vapor compression capacity of the air conditioner. This may lead to downsizing the vapor compression portion of the replacement unit. It is important , however, where the air conditioner on an existing building is being replaced to consider the air delivery capacity of the new unit. A typical 7.5 ton air conditioner supplies air to the space at a rate between 2,300 and 3,700 cfm. Air conditioners smaller than 7.5 ton capacity supply air at a lower cfm. Changing the flow rate may cause significant changes in duct work losses and affect the performance of room air diffusers. Naturally, this limitation may not apply to new construction, where the duct work and diffusers will be designed to be compatible with the air conditioner's supply air flow rate.

Complete rooftop air conditioners and add-on equipment were investigated. Some technologies such as economizers are available as factory installed equipment on available rooftop air conditioners. Others such as heat pipes and condenser pre-coolers are only available as add-ons to a complete packaged rooftop air conditioner. Some such as indirect evaporative cooling modules are available either way. In this case, they were evaluated as part of a complete packaged unit.

Packaged Rooftop Air Conditioner Technologies

Table 1. Packaged Roof Top Air Conditioner Technologies

Technology Incorporated	Equipment Manufacturer	Equipment Model	Energy Efficiency (Note 3)	Estimated Peak kW Demand (Note 4)	Equipment Cost (Note 5)	Advantages	Disadvantages	Comments
Baseline Unit	Trane	YCD090	9.0 EER	10.2	$3,400	Cost	Lowest energy efficiency	Baseline A.C. Unit Single or two compressor models available
Economizer	Trane	YCD090	9.0 EER	10.3	$4,100	Reduced compressor usage at lower ambient air temps	Increased indoor fan kW. Added fan kW if power exhaust needed	1. Controlled by Dry Bulb Temperature Sensor 2. May require power exhaust to control building pressure
Highest Efficiency A.C.	Aaon	RK008 (dual compressors)	11.2 EER	9.2	$6,687		Cost	
A.C. Unit w/ Enthaply Wheel	Aaon	RK008 (dual compressors)	NA	NA	$9,000	Heating energy savings potential	Few hours of large temp diff between outside & exhaust air Moisture transfer for standard wheel	Sensible wheel available
A.C. Unit w/ Indirect Evaporative Cooler	Spec-Air (Indirect Evaporative Cooler w/ Refrigeration)	ACER Model 7.5v or Model 17v	NA	NA	$9,000 for Model 17v; $5,600 for 7.5v	Heating energy savings potential	Trane only at this time Water consumption	Model 7.5v (1 compressor capacity: 4 ton) Model 17v (2 compressors - capacity:7.5 ton)
A.C. Unit w/ Indirect & Direct Evaporative Cooler	Spec-Air (Direct & Indirect Evap Cooler w/ Refrigeration)	ACER Model 7.5v or Model 17v	NA	NA		Heating energy savings potential	Trane only at this time Water consumption	
A.C. Unit w/ Condenser Precooler	Spec-Air		10.1 (estimated EER)	9.3	$1,200	Lowers condensing & suction temps	Water consumptionn	
Indirect Evaporative Cooler Module	De Champs Laboratories	EPX-03	NA	NA	$7,000 for module only w/60% effectiveness	Heating energy savings potential	Compatibility w/ available packaged a.c. not verified. Water Consumptionn	Effectiveness cost: 60%: $2.50/cfm, 70%: $2.75/cfm, 80%: $3.00/cfm
Heat Pipe Heat Exchanger	Circul-Aire	PMW-17TF-66L-11FPI-5R	NA	NA	$8,500	No moisture transfer Heating energy savings potential	Custom made add-on with customized field fabricated duct work	Indirect evaporative cooling of heat pipes available.
Refrigerant Subcooling	FTTS Automatic Controls	NA	NA	NA	$4,000	Stable operation at low head pressures	Only applicable to AC units with thermostatic control valves	

Note 1: All packaged roof top units have nominal capacity of 7.5 tons.
Note 2: Supply air flow rate is assumed to be 2,800 cfm
Note 3: Indicates rated EER in all cases except (estimated). Rated EER is evaluated per ARI Standard 210/240-94 at 95^{o}F db, 75^{o}F wb outdoor entering air and 80^{o}F db, 67^{o}F wb air entering evapoator.
Note 4: Estimated demand is based on EER evaluated at peak load conditions of 102^{o}F db and 70^{o}F wb.
Note 5: Equipment costs do not necessarily include contractor mark up or installation costs.

BASELINE AIR CONDITIONER

Energy and demand savings potential of each technology is based on the difference between the performance of equipment incorporating that technology and that of a baseline air conditioner representative of 7.5 ton air conditioners currently sold in California. Standard efficiency air conditioners manufactured by Trane, York and Carrier are assumed to be representative of those air conditioners. They meet the minimum energy efficiency requirements for a 7.5 ton air conditioner as specified in ASHRAE Standard 90.1. That is, their energy efficiency ratio (EER) is at least 8.9 and, for units capable of capacity reduction, their integrated part load value is at least 8.3. The following information taken from product literature shows the ratings of the standard efficiency air conditioners from these three manufacturers.

Manufacturer	Air Conditioner Model	EER	IPLV
Carrier	48TJ008	8.9	9.35
Trane	YCD090	9.0	9.4
York	DCG090	8.9	9.6

Standard efficiency units sold by these companies incorporate dual compressors as standard equipment. Trane also offers a single compressor unit in the 7.5 ton size at a slightly lower cost which may be more popular where cost is the major factor in a purchasing decision.

These units were tested under conditions specified in ARI Standard 210/240-94. Testing for the EER rating is done at 95°F outside air dry bulb temperature and at 80°F entering air dry bulb temperature and 67°F entering air wet bulb temperature. The IPLV rating is determined from testing at the same entering air conditions and at 80°F outside air dry bulb temperature.

There is a question whether the baseline air conditioner should be defined as being equipped with an air economizer. This investigation assumes that a new 7.5 ton air conditioner could be installed as part of new construction or as a replacement for an air conditioner on an existing building. In some cases, it is necessary to equip the new 7.5 air conditioner with an air economizer to comply with California's Title 24 regulations. First, an economizer is not required if the performance method of compliance is used. When the prescriptive method of compliance is used, the new 7.5 ton air conditioner must be equipped with an economizer for new construction, and, in some situations, for additions and alterations to existing buildings. It is not required when the replacement of the air conditioner on an existing building is considered a repair. Since we can test the units at various inlet air conditions, we can simulate economizer operating conditions without the added complexity of testing this mode of operation. Also, In order to best perform controlled tests on these units, it was decided not to include the air economizer at this time.

TECHNOLOGIES

Several technologies, listed in Table 1, were evaluated more closely in order to determine which technologies should be tested as part of this project. The following is a description of the six most promising technologies:

Economizer

An economizer is a device added to an air conditioner to provide cooling capacity using relatively cool outside air. An economizer consists of an outdoor air damper, relief damper, return air damper, filters, actuator and linkage. Although California regulations may not require a new air conditioner to be equipped with an economizer, most manufacturers supply them as an option. An economizer increases the amount of outside air to be drawn into the unit's evaporator to minimize compressor operation. The amount of cooling capacity available from its operation depends on the outside air temperature and moisture content. The amount of outside air drawn into the unit is often controlled by a probe which senses the outside air dry bulb temperature. The dry bulb sensor is easier to maintain than the enthalpy sensor and is considered adequate for dry climates. However, the maximum energy savings can be obtained from enthalpy sensors. A differential enthalpy control compares the enthalpy of the outside air with that of the return air and modulates the economizer damper to minimize the enthalpy of the mixed air entering the evaporator. An economizer reduces the amount of return air returning to the evaporator. That return air must be exhausted from the building to maintain the building air pressure. Since the indoor fan may have inadequate capacity to do this, a power exhaust fan may be installed to provide this function.

An economizer with a dry bulb sensor may eliminate compressor operation when the outside air temperature is low enough to provide sensible cooling for conditioned spaces. It may also remove latent load from the air conditioner when the moisture content of the outside air is lower than that of the return air normally entering the evaporator coil. When the economizer blades are positioned at the minimum outside air setting most of the return air is mixed with a small percentage of outside air before entering the evaporator. If there is a significant source of moisture in the building, such as a large number of people, the moisture content of the return air could be higher than that of the outside air. Because it represents a larger portion of the air entering the evaporator, the moisture content of the return will significantly affect that of the mixed air entering the evaporator - even when the outside air is relatively dry. An economizer will exhaust a large portion of this moist air to the outside, replacing it with dryer outside air. School buildings and restaurants are good examples of this kind of situation. Some buildings may require lower humidity in the conditioned space. If for example, the humidity is not kept relatively low in a grocery store, a larger de-humidification load is imposed on the cold case refrigeration system instead of the air conditioner. Since the refrigeration system operates at a lower energy efficiency than the air conditioner, operating costs will increase. These are specific examples of increased latent load on the air conditioner. This may be a general benefit for exhausting return air whenever possible, since the air conditioner is typically controlled only by the sensible temperature requirement for the conditioned space. If the latent load is not offset as the compressor operates, the latent load could increase over time.

There are additional energy requirements when a rooftop air conditioner is equipped with an economizer. Adding an economizer to the unit increases the external static pressure on the indoor fan. This increases the fan energy consumption as well as lowering the sensible heat capacity of the unit since heat from the indoor fan affects the supply air temperature. Also, most if not all of the return air must be exhausted from the building when the economizer blades are in the fully open position. The exhaust system must be sized to accommodate this increased air flow to maintain proper building pressure. This can be designed into new buildings, but it must be added to existing buildings increasing electricity consumption. For existing buildings, the building pressure is often relieved using a power exhaust added to the rooftop air conditioner. This power exhaust is mounted close to the unit's outside air intake. This may provide an opportunity for a portion of the exhausted return air to recirculate back into the air conditioner reducing the effectiveness of the economizer.

An economizer could reduce compressor energy consumption substantially at most locations within the PG&E service territory because there are a large number of hours when the outside air temperature is considerably cooler than the design air temperature. This is especially true for buildings which have a large internal load and thus require cooling most of the hours that they are occupied. Economizer damper blades will be at the minimum outside air position when the air conditioner is operating at or near its design condition. Therefore, no demand reduction is expected from an economizer.

No reduction in electrical demand could be realized from the economizer by itself since it operates at cooler outside air temperatures when the air conditioner does not impose its peak demand. Since the emphasis of this project is to investigate technologies that both reduce energy consumption and electrical demand, the economizer by itself does not meet the project's requirement. If, however, a standard efficiency air conditioner is equipped with an economizer and a condenser pre-cooler or a high efficiency air conditioner is equipped with an economizer, the combination of equipment could provide both significant energy consumption and demand reductions. An economizer and a condenser pre-cooler are relatively inexpensive options for a standard efficiency air conditioner as is an upgrade from a standard to high efficiency air conditioner offered by major manufacturers.

Evaporative Cooling

Typically, a relatively small amount of outside air (20% to 30%) is drawn into a commercial building for ventilation purposes. This percentage of outside air minimizes the outdoor air load on the air conditioner. This is especially true at design conditions. When the dry bulb temperature of the outside air decreases naturally a larger quantity of outside air can be drawn into the unit as it is with an economizer. The temperature of the air drawn into the unit can also be reduced and the quantity of outside air increased by evaporative cooling. Equipment now available either pre-cools the outside air using indirect evaporative cooling or a combination of indirect and direct evaporative cooling. With only indirect evaporative cooling the dry bulb temperature of the outside air is reduced without changing its moisture content. Additional cooling of the outdoor air can be accomplished by adding a direct evaporative cooling module downstream of the indirect module. The dry bulb temperature of the air passing through this module will be further cooled, however, its moisture content will be increased

Indirect and indirect/direct evaporative cooling modules are available as stand alone equipment or as part of a package which combines it with mechanical refrigeration. Only one company, Spec-Air, was found that integrates either an indirect or indirect/direct module with mechanical refrigeration into a package rooftop unit with a equivalent capacity of 7.5 tons. Most such modules found were either designed to be used with air handling units or would require customized duct work and control integration to operate with a packaged rooftop air conditioner. De Champs Laboratories model EPX-03 is such a module. Spec-Air has engineered the integration of its indirect and indirect/direct evaporative cooling modules with Trane packaged rooftop air conditioners. The combination is called the ACER.

The ACER is essentially an outside air pre-cooling module attached to the outside air intake of a Trane packaged air conditioning unit. There are two ACER units which have capacities in the 7.5 ton range. The model 7.5v is listed as having a nominal 7.5 ton capacity, however, it incorporates a 3 or 4 ton Trane packaged rooftop air conditioner having an air flow capacity of only 1600 cfm. This unit may be adequate for new construction if its flow rate is compatible with the duct work and diffusers incorporated into the design. It may not be feasible, however, on an existing building since this supply air flow rate may not be within the design range of existing diffusers. The other available model is the 17v which integrates a 7.5 ton Trane packaged rooftop air conditioner with the evaporative cooling module. The supply air flow rate ranges from 2,400 cfm to 3,600 cfm. Therefore, this model would be compatible with either new or existing buildings. The Model 17v ACER can be ordered with a single compressor or a dual compressor 7.5 ton Trane unit. Under most operating conditions, it is likely that only one of the two compressors would operate on the dual compressor model. This would provide capacity reduction capability that would not be available with the single compressor model. Either configuration of the model 17v, however, would provide the full cooling capacity of a Trane 7.5 ton unit in the event the evaporative cooling module fails or is not able to provide the required cooling capacity. This is not the case with the ACER model 7.5v.

This technology has some of the same advantages and disadvantages as the economizer. Fan energy consumption of the indoor fan will increase due to the increased external static pressure. This decreases the sensible capacity of the unit because of the heat gain to the supply air stream from the indoor fan. In addition, the indirect cooling module requires a secondary air fan and a pump to circulate water through an air-to-air heat exchanger. In some cases, the return air from the building can be used as the secondary air stream which may eliminate the need for a power exhaust fan as described for economizers. The moisture content of the outdoor air at locations in PG&E's service territory is relatively low throughout the year and, at times, may be lower than that of the return air. So, indirect evaporative cooling, as with the economizer, may minimize the latent load on the evaporator coil.

As with the economizer there is considerable potential for energy savings throughout the year including during design operating conditions. Since the consumption is lowered even at design conditions, the unit's electrical demand will also be reduced. Note that either a standard efficiency or a high efficiency Trane packaged air conditioner can be specified to be part of the ACER. It is not clear that the incremental cost of a high efficiency packaged air conditioner is justified by the incremental reduction in energy consumption and electrical demand.

High Efficiency Rooftop Air Conditioner

The following table shows the energy performance of high efficiency packaged rooftop air conditioners available from three major manufacturers. It also includes the performance for an 8 ton unit available from Aaon, a smaller manufacturer. Note that there is significant savings potential for the Aaon and Carrier high efficiency unit near design conditions and at part load conditions. The rated EER and IPLV for the standard efficiency air conditioners available from the three major manufacturers averaged about 9.0 and 9.5, respectively.

Manufacturer	Air Conditioner Model	Rated EER	Rated IPLV
Carrier	48HJ008	11.0	11.6
Trane	YCD091D	10.0	10.1
York	D1EG090	10.0	10.5
Aaon	RK008	11.2	12.0

In comparing characteristics of each of the above machines with their standard efficiency counterpart, it was found that manufacturers incorporated a few variations in equipment to obtain a higher efficiency. The most obvious was the increase in heat exchanger surface area. Often they increased the number of rows of the evaporator or condenser while leaving the face area approximately unchanged. One manufacturer replaced a reciprocating compressor with a scroll compressor. Most manufacturers use an orifice to meter the refrigerant into the evaporator, however, one uses a thermostatic expansion valve which may be more efficient over a larger range of temperatures. Table 2 summarizes the information drawn from manufacturers' literature. Note that the information shown for Aaon is standard. Aaon offers a large variety of potential energy saving options. Included are high efficiency indoor fan motors, adjustable speed drive indoor fans, and inlet vanes for volume control on the indoor fan.

Table 2. Comparison of Characteristics of Standard Efficiency and High Efficiency Rooftop Air Conditioners

Manufacturer	Carrier	Carrier	Trane	Trane	York	York	Aaon
Model	48TJ008	48HJ008	YCD090	YCD091	D3CG090	D1EG090	RK008
Nominal Tons	7.5	7.5	7.5	7.5	7.5	7.5	8.0
Efficiency classification	Std.	High	Std	High	Std.	High	High
Rated EER	8.9	11.0	9.0	10.0	8.9	10.0	11.2
Rated IPLV	9.4	11.6	9.4	10.1	9.6	10.5	12.0
Rated air flow (cfm)	2,800	2,800	2,625	2,625	NA	NA	NA
Rated cooling capacity (Btuh)	85,000	90,000	88,000	81,000	NA	NA	NA
Compressor Type	Recip.	Scroll	Recip.	Recip.	NA	NA	Recip.
Evaporator charge (lb/oz/circuit)	4/13/2	8/0/2	6/3/2	6/4/2	6/8/2	9/0/2	8/8/2
Evaporator metering device	Orifice	Orifice	Short Orifice	Short Orifice	NA	NA	TXV
Evaporator coil # rows	3	3	2	3	4	4	2
Evaporator coil fins/in.	15	15	15	15	15	15	12
Evaporator coil face area (sq. ft.)	8	8.9	7.9	7.9	7.8	7.8	11.7
Evaporator nominal fan air flow (cfm)	3,000	3,000	3,000	3,000	NA	NA	NA
Indoor fan type	Centrifugal	Centrifugal	Forward curved centrifugal	Forward curved centrifugal	Centrifugal	Centrifugal	Backward inclined
Condenser coil # rows	1	2	2	2	2	3	2
Condenser coil fins/in.	17	17	16	16	13	13	14
Condenser coil face area (sq. ft)	20.5	20.5	14	14	16.7	16.7	23
Condenser fan air flow (cfm)	6,500	6,500	5,620	5,620	2,900	2,900	2,900
Condenser fan type	Propeller	Propeller	Propeller	Propeller	Propeller	Propeller	Propeller
Condenser fan power (watts)	600	600	NA	NA	NA	NA	900

Heat Wheel and Heat Pipes

Outside air can be cooled by air exhausted from the building as well as by evaporative cooling. Heat wheels and heat pipes are two technologies that could be used to lower the dry bulb temperature of incoming outside air. A heat wheel is a heat exchanger consisting of a rotating cylinder filled with air-permeable medium having a large internal surface area. Adjacent supply and exhaust air streams flow through one-half of the exchanger in a counterflow pattern. Sensible heat is picked up from the hotter stream and stored in the medium and released into the colder stream. Latent heat is transferred by a process of moisture condensation in the more humid air stream and then by evaporation of moisture into the dryer stream. Total (or enthalpy) heat wheels will transfer both moisture and sensible heat from the incoming air into the exhausted air. With relatively dry outside air being drawn into the building, such a heat wheel would likely transfer moisture from the exhausted air into the incoming air. This situation would not be desirable for most small commercial applications, so a sensible heat wheel would be more appropriate. Such an option is available on the Aaon model RK008 air conditioner. The amount of heat transfer is based on the dry bulb temperature difference between the incoming outside air and that of the air exhausted from the building. The Aaon unit has a damper arrangement which causes the return air to be exhausted from the conditioned space.

The relatively large temperature difference between the incoming air and the exhausted return air would cool the outside air significantly at design operating conditions causing a reduction in electrical demand and energy consumption during the hours when the outside air is hot and dry. Energy savings would decrease as the outside air dry bulb temperature approaches the return air dry bulb temperature. The potential for energy saving will depend on the number of hours that the ambient air temperature is 85°F or higher. It's assumed that the return air temperature is typically about 80°F.

A heat wheel integrated into a air conditioner such as the Aaon RK008 would also offer the possibility of heat recovery during the heating season. This would be an advantage where there is a large outside air requirement and a simultaneous heating requirement.

A heat pipe performs the same function as a heat wheel. However, it essentially has no moving parts. It consists of a working fluid permanently sealed in a set of tubes. One end of the tubes, the evaporator section, is in the hotter air stream; while the other, the condenser section, is in the colder air stream. The working fluid absorbs heat in the evaporator section and releases it in the condenser section. A heat pipe is used primarily to transfer sensible heat, however, some heat pipes can transfer moisture. In addition, there is little to no possibility of air leakage from one air stream to another.

Unlike the Aaon's RK008 with the integrated heat wheel, no similar unit was found with an integrated heat pipe. As with evaporative cooling modules, heat pipe modules were found to be applicable to air handling units. One manufacturer provided performance, sizing and price information for a heat pipe module that could be connected to an existing package rooftop air conditioner with field fabricated duct work. The information was based on a dry heat pipe; however, this manufacturer and others make heat pipe modules that are cooled with a water spray to increase their effectiveness.

Condenser Pre-cooler

All manufacturers' performance data shows that an air conditioner's EER improves as the condensing temperature and pressure decreases. The condenser pre-cooler is an evaporative cooling device which lowers the dry bulb temperature of the outside air passing over the condenser coils. The device consists of an evaporative medium, a water reservoir and circulating pump. The condenser fan is used to draw the air first through the medium and then across the condensing coils. As with a direct evaporative cooler, the decrease in the dry bulb temperature is a percentage of the wet bulb depression. The wet bulb depression is the difference between the dry bulb and wet bulb temperature of the outside air. For example, if the dry bulb temperature of the outside air entering the precooler is 105°F and its wet bulb temperature is 75°F, the wet bulb depression would be 30°F. Typically, manufacturers quote 50% to 60% of wet bulb depression to estimate the decrease in outdoor air dry bulb temperature passing through the pre-cooler evaporative medium. Therefore, the dry bulb temperature of the entering air in this example would decrease between 15°F and 18°F as it passes through the precooler.

As the outside air dry bulb temperature decreases, it approaches the wet bulb temperature. Therefore, the wet bulb depression decreases minimizing the effect of the pre-cooler on the condensing temperature. The effect of the condenser pre-cooler on air conditioner energy consumption and electrical demand is greatest at high outdoor dry bulb temperatures.

One pre-cooler manufacturer indicated that this device is most effective on older air conditioners. He said that the condensing temperature on newer machines is already relatively low. In addition, the practical lower limit for the condensing pressure is based on the required differential pressure required to force the refrigerant through the metering device into the evaporator. Some machines have a pressure limit control that maintains a minimum pressure for this purpose.

Electrical demand and some energy consumption reductions are two benefits of the condenser pre-cooler. Some disadvantages are water consumption and maintenance related to water quality. There may also be some small increases in electricity consumed by the condenser fan due to increased pressure drop through the evaporative media.

Based on data from Trane's selection software the reduction in energy consumption and demand is approximately 10 percent for installation of a condenser pre-cooler.

Refrigerant Sub-cooling

Increasing the capacity of the evaporator is one way of increasing the EER of the air conditioner assuming its electrical consumption remains constant. This is what occurs with refrigerant sub-cooling. In an air conditioner without sub-cooling, the temperature of the refrigerant leaving the condenser is slightly below the saturation temperature corresponding to its condensing pressure. Its enthalpy at this condition is the same as its enthalpy as it enters the evaporator. As it passes through the metering device, its pressure and temperature drops to the pressure inside of the evaporator. As the refrigerant moves through the evaporator it vaporizes at essentially a constant pressure and enters the compressor a higher enthalpy. The evaporator's capacity is the difference between the refrigerant's enthalpy as it enters the metering device and that as it leaves the evaporator. This capacity can be increased by decreasing the enthalpy of the liquid refrigerant entering the metering device. That enthalpy can be decreased by sub-cooling the liquid refrigerant. Sub-cooling is a process by which sensible heat is extracted from the liquid refrigerant before it enters the metering device. This can be done by passing it through a liquid-to-air or a liquid-to-liquid heat exchanger.

FTTS Automatic Controls offers an air conditioner modification that incorporates a mini cooling tower external to the air conditioner to sub-cool refrigerant leaving the condenser. The liquid refrigerant leaving the condenser is cooled by passing through a heat exchanger before entering the metering device. It transfers heat to cool water passing through the heat exchanger. That heat is rejected from the water in a small cooling tower. The amount of heat rejection depends on the outside air conditions, increasing with dryer outside air. This arrangement requires electricity to power a pump that circulates the water through the cooling tower and the heat exchanger and to power the cooling tower blower.

This modification is done at the time when the air conditioner is replaced. Essentially, an air conditioner of a given capacity is downsized to a smaller capacity. For example, a 7.5 ton air conditioner would be downsized to nominal 4 ton unit. This would be done by adding the refrigerant heat exchanger and mini cooling tower to a new 4 ton unit. The refrigerant metering device is also downsized and adjusted because the refrigerant pressure decreases when it is sub-cooled. A minimum head pressure is needed make the metering device operate properly.

The energy and demand savings are primarily realized from downsizing the compressor. These savings are offset by the consumption of the cooling tower fan and water circulating pump. Potomac Electric Power Company (PEPCO) has awarded rebates to a small number of its customer for this modification. They do not guarantee that such a modified air conditioner will satisfy a customer's cooling needs.

RECOMMENDATIONS

The above discussion was provided to several individuals within PG&E who have expertise in the HVAC area. They were asked which of the above listed technologies would be the best candidates for further evaluation. In summary, they thought that our testing should characterize the performance of air conditioners equipped with dual compressors and with an evaporative condenser pre-cooler.

Some of these individuals also indicated that a five ton air conditioner was more likely than a 7.5 ton unit to be installed on a small commercial building. Others interviewed during this investigation thought that the most common capacity unit installed is in the range of 7.5 ton to 12 tons. National shipment data from ARI indicates that shipments of unitary 5 ton units far exceed those with a capacity of 7.5 tons. This data, however, may include split systems as well as packaged units and residential as well as commercial. Unitary shipment data from the U.S. Bureau of Census indicates that there are approximately as many 7.5 ton units shipped as 5 ton units. According to ARI, however, it is not possible to apportion national shipment numbers to individual states. So, in order to resolve the question of which capacity unit to evaluate, the advantages of testing a 5 ton unit vs a 7.5 ton unit were considered. After considering the advantages and disadvantages of testing each size, we decided to test a 7.5 ton unit for several reasons. First, the 7.5 ton air conditioner is clearly meant to be installed on a commercial size building. The 5 ton unit is often installed on residential buildings as well as on commercial buildings, so shipment statistics on this size can be misleading. Also, the 7.5 ton unit is most likely to be installed with an economizer, and of the two sizes, is the only one available with dual compressors. Finally, the overall impact of installed 7.5 ton air conditioners on electrical energy consumption and demand was judged to be more significant than that of 5 ton units. In other words, even if there are up to 50% more 5 ton units than 7.5 ton units installed, the overall impact of the 7.5 ton air conditioners is likely to be greater (because of the greater tonnage and corresponding electrical energy use).

Having selected the 7.5 ton capacity air conditioner, we turned to the task of identifying the specific characteristics of the equipment to be tested. A standard efficiency 7.5 ton air conditioner was selected to be the baseline unit. To be representative of those air conditioners currently purchased in the state, the baseline air conditioner selected will be one available from a major manufacturer. Its performance will be the basis for comparison to the performance of air conditioners with more efficient technologies. We selected the single compressor baseline unit, since it may be more likely to be purchased when cost is the dominant purchase criteria as it often is for the small commercial building market. This will also provide a basis for comparing the performance of a high efficiency dual compressor air conditioner with a single compressor standard efficiency air conditioner.

Two higher efficient technologies will be tested. The first will be an evaporative pre-cooler installed on the baseline unit condenser. After the baseline unit is tested, the evaporative pre-cooler will be added on the condenser. The second technology is a high efficiency air conditioner with dual compressors. Plans are to select one from a major manufacturer with the highest EER and IPLV rating.

We have decided not to equip either test unit with an economizer. We believe that the performance of economizer is related to the proper functioning of its components and as such is more of a test of equipment reliability than energy saving potential. In addition, we can determine the potential energy savings from a properly operating economizer in our testing laboratory without actually installing one on either the baseline or high efficiency test unit. Finally, both test units will be equipped with a natural gas heater as well as cooling capacity, since this configuration is thought to be typical for most commercial installations. However, the heating unit will not be tested.

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4. Paul Pieper, Circul-Aire
5. Sal Giglio, Norman S. Wright

6. John Howell, Howell and Associates
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8. Don Felts, Pacific Gas & Electric (PG&E)
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APPENDIX II

Task 4 Interim Report
Test and Evaluation Plan

Evaluation of Small Air Conditioning Units for Northern/Central California Climates

Task 4 -Develop Test and Evaluation Plan

CEC Contract # 500-97-10

Project #1

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Executive Summary

This report describes the detailed plan for testing and evaluating the packaged rooftop air conditioning technologies described and selected in Task 3. This includes identification of the performance indicators to be determined, the test procedures needed to accurately determine these indicators, and the required test facility modifications needed to perform these tests.

This Task was divided into several sub-tasks in order to develop the test and evaluation plan. These sub-tasks included:

- Research current testing standards and other test facilities
- Determine how the test results may be used to enhance performance analysis models
- Define the characteristic performance parameters
- Develop a specific test procedure
- Specify the test instrumentation and its required level of accuracy
- Design the modifications to the test facility

Standards and literature from The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), The Air-Conditioning and Refrigeration Institute (ARI), and others were reviewed for information on relevant testing methods. The DOE-2 building energy analysis program was investigated as the potential tool for using the test results to predict overall performance of these technologies in selected applications. This led to the definition of the required performance parameters from the testing. These will include:

- Cooling Capacity (total, sensible, and latent)
- Indoor-side air flow rate versus static pressure
- Electric power demand and energy use
- Energy efficiency ratio (EER) and Integrated Part Load Value (IPLV)
- Degradation coefficient (for cycling performance)
- Evaporative pre-cooler effectiveness and water consumption (if applicable)

Once the performance parameters were defined, a preliminary uncertainty analysis was performed to evaluate instrumentation accuracy requirements. The test standards mentioned above also gave information about required measurement uncertainties. Based on all of the above information, a general test procedure was developed.

The testing will use the Indoor Air-Enthalpy Method as the primary means of performance testing. As a secondary measurement, the Outdoor Air-Enthalpy Method will also be used. Testing is planned over outdoor conditions ranging from 55 ^{o}F - 115 ^{o}F dry bulb temperature, and indoor conditions ranging from 57 ^{o}F - 75 ^{o}F wet bulb temperature (at 80 ^{o}F dry bulb). Other combinations of temperatures are also planned for special cycling tests and for testing of units equipped with evaporatively cooled condensers.

Starting with the existing test facility at PG&E's Technology Center in San Ramon, modifications required to meet the test plan described above were identified. The work required to modify the facility was put out to bid and awarded. The design of the outdoor and indoor test rooms will allow testing to the procedures described above. This design is described further in the body of the report. Work is currently underway on the test facility modifications.

INTRODUCTION

The manufacturers of packaged rooftop air conditioners design their systems to meet the needs of the largest market segment. This means a design that will operate well in the humid climate of the eastern and southern United States. Like most of the southwest part of the country, California usually has a dry climate during the hot summer months, and there are a number of opportunities for efficiency improvement that are not adequately utilized. Some examples include evaporative pre-cooling of the condenser air, and taking advantage of the lower latent load of the outdoor air for ventilation. Also, dry air retains less heat, which results in a larger temperature change between day and night, and provides for better performance of load shifting thermal storage systems. By taking advantage of these opportunities, the power demand and energy usage for air conditioning can be reduced, resulting in savings for both the customer and the utility.

The goal of this project is to examine the potential for efficiency improvement of packaged rooftop air conditioners in California's hot/dry climate. In order to determine their relative performance, or the effectiveness of performance enhancing devices, a standard plan for testing must be created. This plan will establish the requirements for a testing facility and instrumentation, specify the test conditions for the air conditioning equipment, and outline the test procedures to be followed. By establishing a plan and adhering to its provisions, the results obtained from a test program will be assured to be useful and accurate.

This report documents the methodology followed and the resulting test plan created. The task of developing the test plan was divided into several sub-tasks. These sub-tasks included:

- Research current testing standards and other test facilities
- Determine how the test results may be used to enhance performance analysis models
- Define the characteristic performance parameters
- Develop a specific test procedure
- Specify the test instrumentation and its required level of accuracy
- Design the modifications to the test facility

Each of these tasks is described in detail in separate sections of this report.

STANDARD TEST METHODS

Standard methods for testing air conditioning systems have been established by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), the Air-Conditioning and Refrigeration Institute (ARI), and the Department of Energy (DOE). Several of these have been reviewed and incorporated into the test plan, and these are summarized below.

ANSI/ASHRAE Standard 37-1988: Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment

ASHRAE 37 is the primary standard for establishing test methods for determining the cooling capacity of air-conditioning systems. It is applicable to units rated up to 135,000 Btu/h (11¼ tons). It defines the performance characteristics and related measurements, testing methods, facility requirements, and the required level of accuracy and test tolerances for instrumentation. It does not specify the test conditions (air temperatures), and is limited to only steady-state testing. The defined performance characteristic is the cooling capacity (sensible, latent, and total), as determined by various methods.

ANSI/ASHRAE Standard 116-1983: Methods of Testing for Seasonal Efficiency of Unitary Air-Conditioners and Heat Pumps

ASHRAE 116 is only applicable to units rated up to 65,000 Btu/h (5.4 tons). It expands on ASHRAE 37 to specify standard test conditions and to provide methods of testing under cyclic system conditions. From the cyclic testing, the standard defines the performance characteristics of degradation coefficient (C_D) and seasonal energy efficiency ratio (SEER). It does not specify the cycling interval, but references the interval specified in ARI 210/240.

ARI Standard 210/240-94: Standard for Unitary Air-Conditioning and Air-Source Heat Pump Equipment

ARI 210/240 incorporates and builds on the provisions of ASHRAE 37 and 116. Conditions are established for rating systems with capacities up to 135,000 Btu/hr. It also includes part load testing for units with multiple or variable speed compressors, and the definition of an integrated part load value (ILPV) as a single value representing part-load performance. The cyclic performance testing of systems rated up to 65,000 Btu/hr from ASHRAE 116 is included and expanded on, and a single cycling interval is established (6 minutes on and 24 minutes off). One difference is that ASHRAE 116 requires the indoor-side fan to run continuously when the system is cycled off, whereas ARI 210/240 allows the indoor-side fan to cycle on and off as governed by the system's automatic controls.

ARI Standard 340/360-93: Standard for Commercial and Industrial Air-Conditioning and Heat Pump Equipment

ARI 340/360 extends the range of ARI 210/240 to include systems rated between 135,000 and 250,000 Btu/hr (11¼ to 20.8 tons). Methods for testing units incorporating outdoor air coils are provided.

US Department of Energy Code of Federal Regulations, Title 10, Chapter 11, Part 430, Subpart B, Appendix M: Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners

DOE 10 CFR 11 is essentially identical to ARI 210/240, involving the same testing methods, procedures, and conditions.

Other supplemental standards are referenced by the test standards listed above for the purpose of establishing standard measurement methods. These were also reviewed and include:

- Air Flow Rate:
  ANSI/ASHRAE Standard 41.2-1987: Standard Methods for Laboratory Airflow Measurement
  ANSI/ACMA 210-85 / ANSI/ASHRAE Standard 51-1985: Laboratory Methods of Testing Fans for Rating
- Temperature
  ANSI/ASHRAE Standard 41.1-1986: Standard Method for Temperature Measurement
- Humidity
  ANSI/ASHRAE Standard 41.6-1994: Method for Measurement of Moist Air Properties

TESTING FACILITIES

The test standards only go so far to describe the requirements for the testing apparatus. Additional insight was obtained by examining what others have done in constructing their own test facilities. A search conducted through PG&E's Pacific Energy Center and through the Internet identified a small number of test facilities or manufacturers of testing apparatus, and these are listed below. These cover a range of categories, including academic and government research laboratories, independent test service providers, HVAC equipment manufacturers, and test chamber manufacturers. Some of these were also contacted by phone for additional information.

- **Energy Systems Laboratory, Texas A&M University (College Station, TX).** Two side-by-side psychrometric chambers for controlled temperatures and humidities. Can accommodate equipment up to 10-ton capacity, and measure air flow rates from 150 to 5,000 CFM.
- **Ray W. Herrick Laboratories, Purdue University (West Lafayette, IN).** Their psychrometric chambers were upgraded in 1996. The facility is capable of controlling the dry bulb temperature within the range of -20 to 130°F, the dew point temperature within 20 to 100°F, and can handle equipment up to 10-tons.
- **Larson Building Systems Laboratory, Joint Center for Energy Management, University of Colorado (Boulder, CO).** Not a facility for testing packaged systems, but rather a HVAC simulator for evaluating controls and components.
- **Air Conditioning and Refrigeration Center, University of Illinois at Urbana-Champaign.** Psychrometric rooms for unitary/split A/C systems.
- **Thermal Test Facility, National Renewable Energy Laboratories (Golden, CO).** Psychrometric chambers intended primarily for the testing of desiccant dehumidifiers and other efficiency enhancing components.
- **Thermal Technology Centre, National Research Council (Ottawa, Ontario, Canada).** Two controlled-environment chambers for testing residential and light commercial scale heat pumps, air-conditioners, and refrigeration equipment up to 10-ton capacity. Indoor room is 16.4' x 13.1' x 7.5' high and operates from 41 to 140°F. The outdoor room is 24.9' x 13.1' x 9.8' high, and can operate from -31 to 140°F and 10 to 100% humidity.
- **Lennox Industries R&D Center for Air Conditioning Research (Carrollton, TX).** Facility for testing and rating their product line of air conditioners.
- **Automated Test Labs (Philadelphia, PA).** The psychrometric lab can provide independent performance testing of unitary, split system, or rooftop HVAC equipment with capacities up to 75-tons and airflow rates up to 40,000 CFM. Four test facilities covering different ranges of equipment capacity.
- **Environmental Tectonics Corporation (Southampton, PA).** Custom designed psychrometric test rooms and code testers (airflow and enthalpy measurement systems) for HVAC and automotive industry.
- **Tescor, Inc. (Warminster, PA).** Manufacturer of HVAC psychrometric test rooms for industry.

- **Digital Interface Systems, Inc. (Benton Harbor, MI).** Designs, builds and installs state-of-the-art psychrometric / environmental test facilities for HVAC equipment manufacturers. Facilities have varied from 1 to 30-tons capacity.
- **Ransco Industries (Oxnard, CA).** Manufacturer of psychrometric test facilities for major HVAC component and system manufacturers. Facilities designed for nominal capacities from 1 to 100-tons. Primarily markets to the automotive industry.

PERFORMANCE ANALYSIS METHODS

One of the outcomes of this test program can be to provide building energy analysis computer models with a better estimate of HVAC system performance at off-rated design conditions. This would allow for greater accuracy in predicting the performance of the efficiency-enhanced equipment when installed.

DOE-2 is the most commonly used computer model for building energy usage prediction. In this model, default curves are provided to calculate the adjustment factors that are used to modify the design capacity at the ARI standard conditions (indoor side inlet: 80°F dry-bulb, 67°F wet-bulb; outdoor side inlet: 95°F dry-bulb) to the actual operating conditions. Most of the adjustment factors are functions of only the indoor-side entering wet-bulb temperature (EWB) and the outdoor-side inlet dry-bulb temperature (ODB). Other corrections are applied for off-design indoor-side airflow rate and part load ratio (PLR).

The parameters that can be changed in the DOE-2 model provide direction for a testing program. Thus, the testing shall be oriented towards obtaining values or functions for these parameters, as applicable. Some of these are listed below, using the keywords used by DOE-2.

Performance Parameters Required at Design Conditions (EWB=67, EDB=80, ODB=95):

- COOLING-CAPACITY: total cooling capacity, excluding indoor fan power
- COOLING-EIR: electric input ratio (3.413/EER), excluding indoor fan power
- COOL-SH-CAP: sensible cooling capacity
- COIL-BF: coil bypass factor

System Characteristics:

- COMPRESSOR-TYPE: Single or two speed
- OUTSIDE-FAN-MODE: Continuous or intermittent (i.e. only when compressor is operating)
- OUTSIDE-FAN-KW: Amount of electrical energy used to operate the outside fan.
- OUTSIDE-FAN-T: ODB below which the condenser fans will not operate
- CRANKCASE-HEAT: amount of electrical energy used to heat the crankcase when the compressor is off
- CRANKCASE-MAX-T: ODB temperature above which the crankcase heater will not operate
- MIN-UNLOAD-RATIO: fraction of full load below which compressor unloading stops and hot gas bypass or cycling begins

- MIN-HGB-RATIO: fraction of full load below which hot gas bypass stops and unit cycling begins

Adjustment Factors:

- COOL-CAP-FT: Adjusts total cooling capacity, function of EWB, ODB
- COOL-EIR-FT: Adjusts electric input ratio, function of EWB, ODB
- COOL-EIR-FPLR: Adjusts electric input ratio, function of PLR (only applies to compressors that can be unloaded)
- COOL-SH-FT: Adjusts sensible cooling capacity, function of EWB, ODB
- RATED-CCAP-FCM: Adjusts total cooling capacity, function of airflow rate ratio
- RATED-CEIR-FCFM: Adjusts electric input ratio, function of airflow rate ratio
- RATED-SH-FCFM: Adjusts sensible cooling capacity, function of airflow rate ratio
- COIL-BF-FT: Adjusts coil bypass factor, function of EWB, ODB
- COIL-BF-FCFM: Adjusts coil bypass factor, function of airflow rate ratio
- COOL-FT-MIN: minimum ODB for accurate interpolation of adjustment curves

CHARACTERISTIC PERFORMANCE PARAMETERS

The review of the testing and rating standards, other test facilities, and the adjustable parameters of modeling tools provided a list of performance parameters to be determined through this test program. Other parameters were added as deemed necessary to determine the performance of add-on energy-enhancing devices. These parameters are listed below, with descriptions of the measurements necessary to determine them.

- **Cooling Capacity** (Total, Sensible, and Latent)
  The primary method for determining capacity is the indoor-side air enthalpy method, which requires measurements of airflow rate through the coil, and inlet and outlet dry-bulb temperatures and humidity (or wet-bulb temperature). As a secondary measurement for comparison, the cooling capacity will also be calculated from similar measurements on the outdoor coil, less the power demand. Capacity adjustment factors will be determined as functions of indoor-side entering wet-bulb temperature, outdoor-side entering dry-bulb temperature, indoor-side air flow rate, and part load operation.

- **Electric Demand**
  The total demand of the system will be measured simultaneously with the measurements for capacity. As possible, the partial demands of the unit sub-systems will also be measured.

- **Energy Efficiency Ratio** (EER)
  EER is the ratio of capacity to demand, as averages or totals over a specific period. The EER determined as a function of part load operation is used to determine an integrated part load value (IPLV) and an adjustment factor for EER.

- **Coil Bypass Factor**
  Number representing the portion of the supply air that does not reach the coil surface condition. Used to adjust the sensible cooling capacity and sensible EER.

- **Supply Airflow Rate** (as a function of external resistance)
  The airflow rate is measured from the pressure drop across a bank of nozzles, the total area

of the nozzles, and measurements to determine the air density. The flow rate is trended as a function of the external resistance (or the differential pressure between the supply and return). Measurements of capacity and EER at various flow rates may be used to determine the adjustment factors for each.

- **Degradation Coefficient** (C_D)
  Parameter describing the decrease in performance due to unit cycling. Determined through two dry-coil tests: steady state and cycling. Once determined, the parameter may be used to calculate a Seasonal Energy Efficiency Ratio (SEER).
- **Evaporative Pre-cooler Effectiveness and Water Consumption** (if so equipped)
  Effectiveness is determined from measurements of inlet dry- and wet-bulb temperatures and outlet dry-bulb temperature.

TEST PROCEDURE DEVELOPMENT

The information contained in the reviewed rating standards was compiled and combined to produce a test plan specific to achieving the desired results of this project. The final test plan is included with this report as **Attachment A**. This test plan was intended to be a stand-alone document, which would not require having to reference the original standards. Thus, much of the information contained within it was transferred directly from the standards, which are only referenced when there is more detail to be found within them. The requirements for testing methods and apparatus, instrument accuracy and measurement tolerances remain the same. The plan is then a merging of the various sources of testing methods and requirements, and not a straight duplication of any one of them. The requirements of the standards were added to where additional information was required for the purposes of this test program. In particular, additional tests were added for the sensitivity of the performance parameters to a wider range of operating conditions, more cycling modes, and evaluation of add-on components.

The plan begins with statements regarding its purpose and scope: why the plan was developed and to what systems it applies. This is followed by a table of definitions for different parameters referenced throughout the plan. The performance indicators, as listed above, are then described along with the methods by which they will be determined. The next sections describe the requirements for the test facility and the measuring instruments. The plan then goes into describing the methods of testing, including the ARI standard rating tests, calculation of SEER and ILPV, additional sensitivity tests to determine the DOE-2 adjustment factors, cyclic performance, variable airflow versus external resistance, and evaporative pre-cooler effectiveness. The procedures to be followed during the tests are described, as is the data to be recorded and the tolerances for key measurements through the course of a test. Finally, the test standards from which the plan was derived are referenced.

The test plan as it exists serves as the basis from which the tests will be run. There may be certain test conditions for specific test units which can not be realistically obtained. Deviations from the test plan will be duly noted in any test results. Additional tests may be added so long as the basic requirements of the original standards are followed.

TEST FACILITY MODIFICATIONS

The planned test facility will utilize two existing structures at PG&E's Technology Center in San Ramon. These were originally constructed in the late 1970's for the purpose of evaluating passive solar space heating technology. One of the two structures was modified in 1995 to serve as the controlled environment space for evaluating residential air conditioners. The structure had two rooms: a control room for the data collection, and a chamber that was supplied with heated or humidified outdoor air to create a range of environmental conditions around the condensing unit. The discharge from the unit under test was captured and exhausted from the chamber. The load for the system was provided by a length of duct external to the chamber, which supplied heated outside air to the evaporator coil. The limitation of this system was that tests could only be conducted when the ambient temperature and humidity were less than what was required by either the chamber or the load duct. The system was adequate for testing residential sized units. Its limitations only affected the testing schedule.

The residential system testing greatly influenced the planned system modifications that would be required to expand the facility for testing the larger capacity commercial roof-top air conditioners. Other influences were the requirements outlined in the testing standards, and the researched descriptions of other testing facilities. Most of the modifications are centered around the previously unused second building, which has a single room and can accommodate a physically larger unit. There are three major planned modifications for this building. First, the roof will be raised to increase the volume of the chamber. This is being done as per a standard requirement to provide sufficient clearance above the discharge of the unit, and to increase the working space. The larger volume will also help stabilize the air conditions around the test unit. The roof modification will eliminate a beam and center post that disrupts the floor space. The second modification is to provide an access passage to move the test units in and out of the chamber. An existing wall of windows will be replaced with a double door, which will be insulated and tightly weather stripped. The final and most significant modification is the installation of the space conditioning system for this room. This equipment (20 ton heat pump, electric heater, and humidifier) will be located on a slab behind the structure, and will provide cooling as well as heating and humidification. The system will allow for air from the chamber to be recirculated and conditioned, or 100% outside air may be used as before, or a mixture of the two.

A sketch of the modifications to this building is shown in **Figure 1**. The goals are to provide a larger volume of air to meet the requirements of the test unit, keep the conditions of the room stable through the course of a test, and be able to conduct the tests over a wider range of outside conditions. The facility is being designed to handle testing of units up to 10 tons.

The chamber that was used as the outdoor room for the residential system testing will be used as the indoor side room to supply conditioned air for the load on the test unit. The existing blower, electric resistance heater, and humidifier will be reused, although reconfigured to allow for the return of air from the test unit in addition to the use of 100% outside air. In recirculation mode, the indoor side conditioning system only needs to replace the heat and humidity that is removed by the test unit. The space of the chamber acts as a buffer to dampen out fluctuations in the output of the room conditioning system, and to provide stable conditions to be returned to the test unit. The indoor room will be connected to the outdoor room by a pair of ducts, carrying supply and return air to and from the test unit. The layout of the test facility buildings is shown in **Figure 2**.

Figure 1
Modifications to Outdoor Room Structure

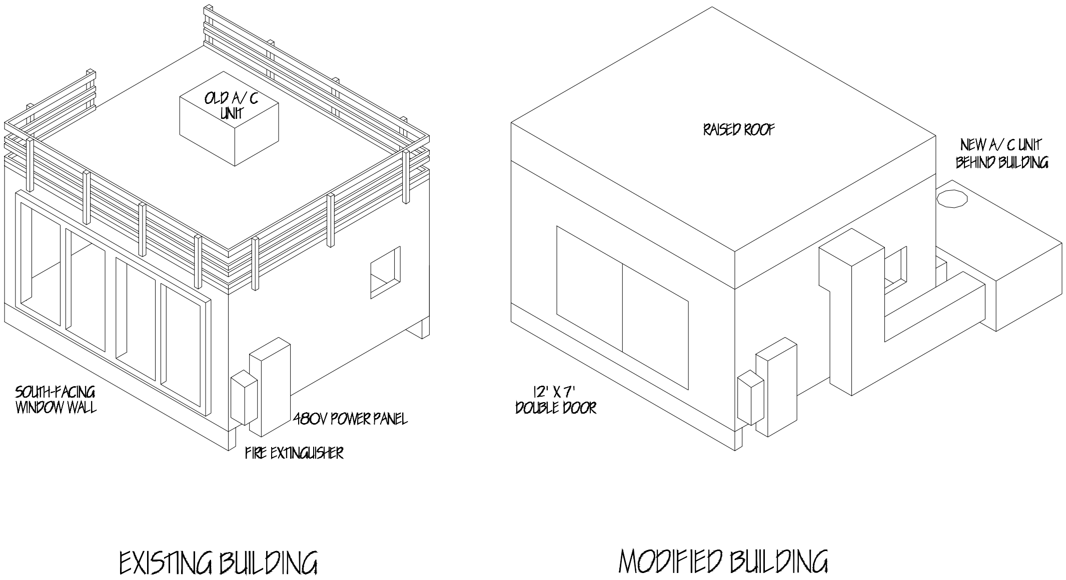
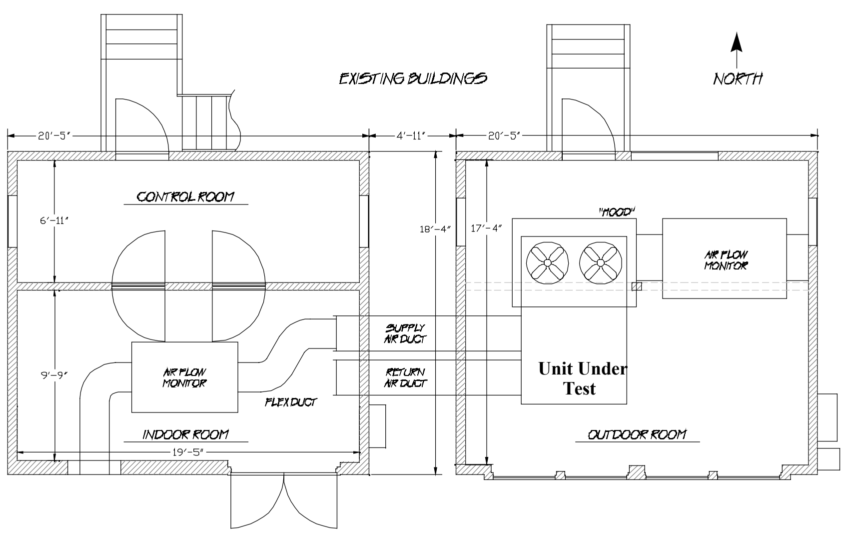


Figure 2
Layout of Test Facility



The requirements for the modifications to the test facility were put into a Request for Proposal (RFP). This RFP was sent out to bid to a number of contractors and eventually awarded. The contractor is required to provide a final design for the modification based on the requirements outlined in the RFP, which will be subject to approval by PG&E. Construction should start soon.

SUMMARY OF TASKS COMPLETED IN THIS PERIOD

- Testing and rating standards and other test facilities were identified and reviewed.

- The adjustable parameters of the DOE-2 modeling program were identified.
- The characteristic performance indicators were defined, and the required measurement accuracy was determined based on a preliminary uncertainty analysis.
- Most of the measurement instruments and the data acquisition and control system were purchased (with PG&E match funds).
- A test plan was developed primarily based on the rating standards.
- The requirements for the modification of the test facility were defined. These were put into an RFP, which was put out to bid and awarded.

NEXT TASK

- Modification of the test facility according to the plan
- Development of the data acquisition and system control program
- Procurement of additional test and control instrumentation, as needed.

Attachment A

Plan for Testing and Evaluation of Packaged Roof-Top Air Conditioning Systems

1. PURPOSE

1.1 The purpose of this test procedure is to provide methods for determining the cooling capacity, electrical demand, and efficiency of roof-top, unitary air conditioning equipment. The procedure is designed to meet the test program goal of determining the sensitivity of these performance indicators to intake air conditions in order to better define the performance adjustment factors used in the DOE-2 modeling program.

1.2 Although this procedure is not intended to establish a performance rating for a particular system, it is necessary to duplicate some of the tests under which it was rated in order to relate the sensitivity of its performance back to its published rating. Thus, most of this test procedure is derived from the rating standards listed in the references. If not explicitly restated in this procedure, the requirements contained in the rating standards shall apply.

2. SCOPE

2.1 This procedure applies to electrically driven, mechanical-compression, unitary air conditioners consisting of one or more assemblies which include an indoor air coil, a compressor, and an outdoor coil. Where such equipment is provided in more than one assembly, the separated assemblies are designed to be used together.

2.2 This procedure does not include methods of testing the following:

(a) cooling coils for separate use
(b) condensing units for separate use
(c) room air conditioners
(d) heat-operated unitary equipment (i.e. engine driven or absorption chillers)
(e) liquid chilling packages
(f) the heating performance of units which provide both heating and cooling

3. DEFINITIONS

air, standard: dry air at 70°F and 14.696 psi (at these conditions, dry air has a mass density of 0.075 lb/ft^3).

apparatus: refers exclusively to test room facilities and instrumentation.

ARI: Air-conditioning and Refrigeration Institute.

ASHRAE: American Society of Heating, Refrigeration and Air-conditioning Engineers, Inc.

capacity: the rate, expressed in Btu/h, at which the equipment removes heat from the air passing through it under specified conditions of operation.

coil, indoor: the heat exchanger which removes heat from the conditioned space (evaporator).

coil, outdoor: the heat exchanger which rejects heat to a source external to the conditioned space (condenser).

continuously recorded: a method of recording measurements in intervals no greater than 5 seconds.

cooling load factor (CLF): the ratio of the total cooling done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady-state cooling done over the same period at constant ambient conditions.

cyclic test: a test where the indoor and outdoor conditions are held constant, but the unit is manually turned "on" and "off" for specific time periods to simulate part-load operation.

degradation coefficient (C_D): the measure of the efficiency loss due to the cycling of the unit.

dry-coil test: a test conducted at an intake wet-bulb temperature and a dry-bulb temperature such that moisture will not condense on the evaporator coil of the unit (i.e. the dew-point temperature of the inlet air is less than the evaporator temperature).

energy efficiency ratio (EER): a ratio calculated by dividing the cooling capacity in Btu/h by the power input in watts at any given set of rating conditions, expressed in Btu/Wh.

equipment: refers exclusively to the unitary equipment to be tested.

indoor side: that part of the system which removes heat from the indoor air stream.

integrated part load value (IPLV): a single representative number for efficiency at partial loading.

latent cooling: the amount of cooling in Btu's necessary to remove water vapor from the air passing over the indoor coil by condensation during a period of time. (Energy equivalent to the reduction of moisture without a change in temperature.)

outdoor side: that part of the system which rejects heat to a source external to the indoor air stream.

part load factor (PLF): the ratio of the cyclic energy efficiency ratio to the steady-state energy efficiency ratio at identical ambient conditions.

published rating: a statement of the assigned values of those performance characteristics, under stated rating conditions, by which a unit may be chosen to fit its application. These values apply to all units of like nominal size and type (identification) produced by the same manufacturer. The term "published rating" includes the rating of all performance characteristics shown on the air-conditioner or published in specifications, advertising or other literature controlled by the manufacturer at stated rating conditions.

rating conditions: any set of operating conditions under which a single level of performance results, and which causes only that level of performance to occur.

refrigerant, volatile: one which changes from the liquid to the vapor state in the process of absorbing heat.

seasonal energy efficiency ratio (SEER): the total cooling of a central air-conditioner in Btu's during its normal usage period for cooling divided by the total electric energy input in watt-hours during the same period.

sensible cooling: the amount of cooling in Btu's performed by a unit over a period of time, excluding latent cooling. (Energy equivalent to the reduction of air temperature without a change in moisture.)

shall: where "shall" or "shall not" is used for a provision, that provision is mandatory.

should, recommended, or "it is recommended": "should", "recommended", or "it is recommended" is used to indicate provisions which are not mandatory, but which are desirable as a good practice.

single package unit: any central air conditioner in which all the major assemblies are enclosed in one cabinet.

split system: any central air conditioner in which one or more of the major assemblies are separate from the others.

steady-state test: a test in which all indoor and outdoor conditions are held constant within prescribed tolerances and the unit is in a non-changing operating mode.

test condition tolerance: the maximum permissible variation of the average of the test observations from the standard or desired test condition.

test operating tolerance: the maximum permissible difference between the maximum and the minimum instrument observation during a test.

wet-coil test: a test conducted at an intake wet-bulb and a dry-bulb temperature such that moisture will condense on the test unit evaporator coil (i.e. the entering dew-point temperature is above the evaporator temperature).

will: is used to indicate provisions specific to this test procedure, whether or not derived from a rating standard.

4. PERFORMANCE INDICATORS

4.1 The following indicators are used to describe the performance of an air conditioning system.

(a) Total cooling capacity, Btu/h
(b) Sensible cooling capacity, Btu/h
(c) Latent cooling capacity, Btu/h
(d) Indoor-side air flow rate, SCFM
(e) Indoor-side external resistance to air flow, inches of water
(f) Total power input to equipment or power inputs to all equipment, watts
(g) Energy efficiency ratio
(h) Degradation coefficient (C_D)
(i) Evaporative pre-cooler effectiveness and water consumption (if so equipped)

4.2 Capacity

4.2.1 The primary method for determining the capacity of a system is the *Indoor Air-Enthalpy Method*. In this method, the capacity is determined as the air enthalpy difference across the evaporator coil multiplied by the corresponding air mass flow rate. The entering and leaving air enthalpies are determined from measurements of dry- and wet-bulb temperature (or humidity) and pressure.

4.2.1.1 Total, sensible, and latent indoor cooling capacities based on the indoor air-enthalpy method are calculated by the following equations:

$$q_{tci} = 60\, Q_{mi}\, (h_{a1} - h_{a2}) / [v'_n\, (1 + W_n)] \qquad \text{Total Cooling}$$

$$q_{sci} = 60\, Q_{mi}\, c_{pa}\, (t_{a1} - t_{a2}) / [v'_n\, (1 + W_n)] \qquad \text{Sensible Cooling}$$

$$c_{pa} = 0.24 + 0.444\, W_n$$

$$q_{lci} = 63{,}600\, Q_{mi}\, (W_{i1} - W_{i2}) / [v'_n\, (1 + W_n)] \qquad \text{Latent Cooling}$$

4.2.2 Alternative measurements of capacity shall be made for comparison with the indoor air-enthalpy method. The alternate method shall agree with the primary method to within 6.0% to constitute a valid test. Alternate methods include one or more of the following:

4.2.2.1 *Outdoor Air-Enthalpy Method:* In this method, the heat rejected from the condenser is determined from measurements of the outdoor air enthalpy difference through the condenser and the corresponding air mass flow rate. The system capacity is then determined by subtracting the electrical energy input (converted to Btu/h) from the heat rejected, as shown in the following equation:

$$q_{tco} = 60\ Q_{mo}\ (h_{a4} - h_{a3}) / [v'_n\ (1 + W_n)] - 3.41\ E_t$$

or for air-cooled equipment which does not re-evaporate condensate or operate with an evaporative pre-cooler,

$$q_{tco} = 60\ Q_{mo}\ c_{pa}\ (t_{a4} - t_{a3}) / [v'_n\ (1 + W_n)] - 3.41\ E_t$$

4.2.2.2 *Refrigerant Enthalpy Method:* In this method, capacity is determined from the refrigerant enthalpy change through the evaporator and its flow rate. Enthalpy changes are determined from measurements of entering and leaving pressures and temperatures of the refrigerant, and the flow rate is determined by a suitable flow meter in the liquid line.

This method may be used for tests of equipment in which the refrigerant charge is not critical and where normal installation procedures involve the field connection of refrigerant lines or charging of the system. This method shall not be used for tests in which the refrigerant liquid leaving the flow meter is sub-cooled less than 3°F, nor for tests in which the superheat of the vapor leaving the indoor section is less than 5°F.

The refrigerant flow rate shall be measured by an integrating-type flow meter connected in the liquid line upstream of the refrigerant expansion valve. This meter shall be sized so that its pressure drop does not exceed the vapor pressure change that a 3°F temperature change would produce. Temperature and pressure measuring instruments and a sight glass shall be installed immediately downstream of the meter to determine if the refrigerant liquid is adequately sub-cooled. Sub-cooling of 3°F and the absence of any vapor bubbles in the liquid leaving the meter are considered adequate. It is recommended that the meter be installed at the bottom of a vertical downward loop in the liquid line to take advantage of the static head of liquid thus provided.

At the end of the test, a sample of the circulating refrigerant and oil mixture may be taken from the equipment and its percentage of oil measured in accordance with the latest issue of ASHRAE Standard 41.4. The total indicated flow rate shall be corrected for the amount of oil circulating.

Total cooling capacity based on volatile refrigerant flow data is calculated as follows:

$$q_{tci} = x\ V_r\ \rho\ (h_{r2} - h_{r1}) - 3.41\ E_i$$

4.2.2.3 *Cooling Condensate Method:* By collecting and measuring the amount of water condensed by the evaporator over a period of time, the average latent cooling capacity over the same period may be determined. The latent cooling capacity is calculated as follows:

$$q_{lci} = 1061\ w_c$$

4.3 Demand

4.3.1 The demand of a system is the total electrical power input to the unit, which includes the power used by the compressor(s), condenser fan, evaporator fan, and controls.

4.4 Efficiency

4.4.1 The energy efficiency ratio is the capacity of the unit divided by its electrical demand under the same operating conditions, expressed in Btu/Wh.

5. APPARATUS

5.1 The a diagram of the test facility arrangement is shown in Attachment 1. An air flow measuring device is attached to the equipment air discharge (evaporator and/or condenser) in each room. The outlet from this device is directed towards suitable reconditioning equipment. The discharge air from the reconditioning apparatus provides the desired equipment intake temperature and humidity

5.2 The indoor-side test room may be any room or space in which the desired test conditions can be maintained within the prescribed tolerances. It is recommended that air velocities in the vicinity of the intake to the equipment under test not exceed 500 fpm. The test installation shall be designed such that there will be no air flow through the cooling coil due to natural or forced convection while the indoor fan is "off". This may be accomplished by installing dampers upstream and downstream of the test unit to block the "off" period air flow.

5.3 The equipment to be tested is typically located in the outdoor-side test room. This room shall be of sufficient volume and shall circulate air in a manner such that it does not change the normal air circulating pattern of the equipment under test. It shall be of dimensions such that the distance from any room surface to any equipment surface from which air is discharged is not less than 6 ft and the distance from any other room surface to any other equipment surface is not less than 3 ft, except for floor or wall relationships required for normal equipment installation. The room conditioning apparatus should handle air at a rate not less than the outdoor airflow rate, and preferably should take this air from the direction of the equipment air discharge and return it at the desired conditions uniformly and at low velocities.

6. INSTRUMENTS

6.1 Temperature Measurements

6.1.1 Temperature measurements shall be made in accordance with the latest issue of ASHRAE Standard 41.1.

6.1.2 Sensors shall be calibrated through the measurement system against a secondary standard temperature indicator. Calibration points shall include a water/ice slurry, and a high temperature point which will encompass the range of sensor operation during the testing. The high temperature point may be provided by a heated bath, a hot dry block, or a melting point cell (e.g. gallium). The secondary standard shall have current calibration certification. Record the calibration data on the Temperature Calibration Data Sheet given in Attachment 2, or equivalent. A final calibration check on the instruments shall be performed after testing is complete.

6.1.3 All duct air temperature measurements are to be taken upstream of static pressure taps on the inlet and downstream of the static pressure taps on the outlet.

6.1.4 In-duct temperature measurements shall be taken at not less than three locations at the centers of equal segments of the cross sectional area, or suitable sampling of mixing devices giving equivalent results shall be provided. Ductwork shall be insulated between the place of measurement and the equipment so that heat leakage through the connections does not exceed 1.0% of the capacity.

6.1.5 Indoor-side inlet temperature shall be measured at a minimum of three locations equally spaced over the equipment inlet area, or equivalent sampling means provided. For units without duct connections or enclosure, the temperature measuring instruments or sampling devices should be located approximately 6 in. from the equipment inlet opening or openings.

6.1.6 Outdoor inlet air temperatures shall be measured at locations such that the following conditions are fulfilled:

(a) The measured temperatures shall be representative of the temperature surrounding the outdoor section and simulate the conditions encountered in an actual application.

(b) At the point of measurement, the temperature of air must not be affected by the air discharged from the outdoor section. This makes it mandatory that the temperatures be measured upstream of any recirculation produced.

It is intended that the specified test temperatures surrounding the outdoor section under test shall simulate as nearly as possible a normal installation operating at ambient air conditions identical with the specified test temperatures.

6.1.7 Air velocities over wet-bulb temperature measuring instruments shall be approximately 1000 fpm. Wet-bulb measurements above or below 1000 fpm must be corrected in accordance with ASHRAE Standard 41.1.

6.1.8 For the cyclic, dry coil performance tests, the dry-bulb temperature of the air entering and leaving the cooling coil, or the difference between these two dry-bulb temperatures, shall be continuously recorded. The instruments used should have the following performance:

(a) have a total system accuracy within ±0.3°F of the indicated value.

(b) have a total system response time of 2½ seconds or less. The response time is defined as the time required for the instrumentation to obtain 63% of the final steady state value when subjected to a step change in temperature of 15°F or more, in air.

Cautionary Note: When using temperature transducers in series or parallel (such as thermocouple grids or thermopiles) to obtain the average temperature or average temperature difference in a duct with non-uniform velocities, the overall response time can be much larger than the response time measured at the average velocity.

6.1.9 It is recommended that the same instrumentation be used for making both steady-state and cyclic test measurements.

6.2 Humidity Measurements

6.2.1 As an alternative to, or in addition to, wet-bulb temperature measurements, electronic relative humidity or dew point instruments may be used to determine air moisture.

6.2.2 The accuracy of any humidity instruments used for capacity measurement shall be 1% or better.

6.3 Pressure Measurements

6.3.1 Pressure measurements shall be made with one or more of the following instruments:

(a) inclined water manometer
(b) Bourdon tube gage
(c) electronic pressure transmitters

6.3.2 The accuracy of the pressure measuring instruments shall permit measurements within ±2.0% of the indicated value. Indoor-side external resistance shall be measured with manometers or electronic pressure transmitter(s) having an accuracy of ±0.01 in. of water.

In no case shall the smallest scale division of the pressure measuring instrument exceed 2½ times the specified accuracy

6.3.3 Calibration of Bourdon tube gages or electronic pressure transmitters shall be with respect to a dead-weight tester or by comparison with a liquid column. Calibration shall include a minimum of five points evenly distributed across the range of the instrument. Record the calibration data on the Pressure Calibration Data Sheet given in Attachment 3, or equivalent. Perform a final calibration check on the instruments (zero and full scale) after testing is complete.

6.3.4 One side of the sensor used to measure the indoor-side external pressures shall be connected to four externally manifolded pressure taps in the discharge plenum, these taps being centered in each plenum face at a distance of twice the geometric mean cross sectional dimension from the equipment outlet. If an inlet duct connection is employed, the other side of the pressure sensor shall be connected to four externally manifolded pressure taps centered in each face of the inlet duct; if no inlet duct connection is employed, the other side of the pressure sensor shall be open to the surrounding atmosphere. Inlet and outlet duct connections shall have cross section dimensions equal to those of the equipment and shall be long enough to give accurate readings (i.e. 2½ times the geometric mean of the cross sectional dimensions).

6.3.5 It is recommended that the pressure taps consist of ¼-inch diameter nipples soldered to the outer plenum surfaces and centered over 0.04-inch diameter holes through the plenum. The edges of these holes should be free of burrs and other surface irregularities.

6.4 Air Flow Measurements

6.4.1 Apparatus

6.4.1.1 The air flow measurement apparatus for determining capacity consists of a receiving chamber and a discharge chamber separated by a partition in which one or more nozzles are located. Air from the equipment under test is conveyed via duct to the receiving chamber, passes through the nozzle or nozzles, and is then exhausted to the test room or channeled back to the room conditioning equipment inlet.

6.4.1.2 The nozzle apparatus and its connections to the equipment inlet shall be sealed so that air leakage does not exceed 1.0% of the air flow rate being measured.

6.4.1.3 The center-to-center distance between nozzles in use shall be not less than 3 times the throat diameter of the larger nozzle, and the distance from the center of any nozzle to the nearest discharge or receiving chamber side wall shall not be less than 1½ times its throat diameter.

6.4.1.4 Diffusers shall be installed in the receiving chamber at least 1½ times the largest nozzle throat diameter upstream of the partition wall, and in the discharge chamber at least 2½ times the throat diameter of the largest nozzle downstream of its outlet.

6.4.1.5 A multiple nozzle apparatus shall include access to the nozzles to install or remove covers which prevent air flow through one or more of the nozzles.

6.4.1.6 An exhaust fan, capable of providing the desired static pressure at the equipment outlet shall be installed in one wall of the discharge chamber and means shall be provided to vary the capacity of this fan.

6.4.2 Measurements

6.4.2.1 The static pressure drop across the nozzle or nozzles shall be measured with manometers or electronic differential pressure transmitters having an accuracy of ±1.0% of the reading. One end of the transmitter shall be connected to a static pressure tap located flush with the inner wall of the receiving chamber and the other end to a static pressure tap locate flush with the inner wall of the discharge chamber, or preferably, several taps in each chamber should be manifolded together.

6.4.2.2 Areas of nozzles shall be determined by measuring their diameters to an accuracy of ±0.20% in four places approximately 45 deg. apart around the nozzle in each of two planes through the nozzle throat, one at the outlet and the other in the straight section near the radius.

6.4.2.3 Means shall be provided to determine the air density at the nozzle throat through measurements of dry-bulb temperature, static pressure, and moisture.

6.4.2.4 The throat velocity of any nozzle in use shall be not less than 3,000 fpm, nor more than 7,000 fpm.

6.4.3 Calculations

6.4.3.1 The air flow rate through a single nozzle is calculated by the following equations:

$$Q_{mi} = 1096\ C\ A_n\ (\ p_v\ v'_n\ )^{1/2}$$

or

$$Q_{mi} = 1096\ C\ A_n\ Y\ (\ p_v\ v'_n\ )^{1/2}$$

where Y = expansion factor for the nozzle as given in ASHRAE 51.

$$v'_n = 29.92\ v_n / \{\ P_n\ (\ 1 + W_n\ )\ \}$$

v_n is either calculated or selected from tables in ASHRAE 51 giving specific volume of air at standard barometric pressure for various wet- and dry-bulb temperatures. The discharge coefficient C is obtained from ASHRAE 51 as a function of the nozzle throat Reynolds Number.

6.4.3.2 When more than one nozzle is used, the total air flow rate is the sum of the flow rates of the individual nozzles**.**

6.4.3.3 The flow rate of standard air is calculated as follows:

$$Q_s = Q_{mi} / (\ 0.075\ v'_n\ )$$

6.5 Electrical Instruments

6.5.1 Electrical measurements shall be made with indicating or integrating instruments.

6.5.2 The total system power demand shall be measured with a watt or watt-hour measuring system with an accuracy of ±05% of the indicated value. For non-steady state tests, the total electrical energy shall be integrated over the test period. This applies to both “on” and “off” cycle measurements.

6.5.3 Instruments used for measuring the electrical input to fan motors, compressor motors, or other equipment accessories shall be accurate to ±2.0% of the indicated value.

6.5.4 Voltages shall be measured at the equipment terminals.

6.6 Volatile Refrigerant Flow Measurement

6.6.1 If performed, volatile refrigerant flow shall be measured with an integrating type flow meter having an accuracy of ±1.0% of the indicated value.

6.6.2 This meter shall be sized so that its pressure drop does not exceed the vapor pressure change that a 3°F temperature change would produce.

6.7 Liquid Flow Measurement

6.7.1 Water flow rates (such as the supply to an evaporative pre-cooler) shall be measured with a liquid flow meter or quantity meter having an accuracy of ±1.0% of the indicated value.

6.7.2 Condensate collection rates shall be measured with a liquid quantity meter measuring either weight or volume and having an accuracy of ±1.0% of the indicated value.

6.8 Speed Measuring Instruments

6.8.1 Speed measurements (i.e. fans) shall be made with a revolution counter, tachometer, stroboscope, or oscilloscope having an accuracy of ±1.0%.

6.9 Time and Weight Instruments

6.9.1 Time measurements shall be made with instruments having an accuracy of ±0.20% and weight measurements apparatus having an accuracy of ±0.2%.

7. METHODS OF TESTING

7.1 The tests to be conducted fall under two categories:

(a) Tests done under the same conditions under which the equipment was rated to establish a reference basis.

(b) Tests to determine the variation in the performance parameters as a function of inlet air conditions.

7.2 Rating condition tests

7.2.1 The conditions for the standard rating tests are listed in Table 1. Two steady-state wet coil tests are performed: Tests A and B. Test A and Test B shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80 F and a wet-bulb temperature of 67 F. Test A is to be conducted at an outdoor dry-bulb temperature of 95 F, and Test B at 82 F. For those units which reject condensate to the condenser, located in the outdoor side of the unit, the outdoor wet-bulb temperature surrounding the outdoor side of the unit shall be 75 F in Test A and 65 F in Test B.

7.2.2 The test requirements for two speed compressor units, two compressor units, or units with cylinder unloading are that Test A and Test B shall be performed at each compressor speed or at each compressor capacity.

7.2.3 Tests C and D are optional tests to be conducted when cyclic performance parameters are to be measured in order to determine the degradation coefficient, C_D, and the seasonal energy efficiency ratio (SEER). These tests are performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80 F and a wet-bulb temperature which does not result in formation of condensate on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 F or less be used.) The dry-bulb temperature of the air entering the outdoor portion of the unit shall be 82 F in both tests. Test C is a steady-state dry coil test to determine a reference performance. Test D shall be conducted by cycling the unit "on" and "off" by manual or automatic operation of the normal control circuit of the unit. The indoor fan shall also cycle "on" and "off"; the duration of the indoor fan "on" and "off" periods being governed by the automatic controls which the manufacturer normally supplies with the unit.

7.2.4 Test standards call for two tests at extremes of operation, intended to ensure that the equipment will operate under these extremes and are normally not used for rating purposes. These include a maximum condition test at an outdoor dry-bulb temperature of 115 F, and a low temperature cooling test at an outdoor and indoor condition of 67 F dry-bulb and 57 F wet-bulb.

7.2.5 Partial load tests to determine the ILPV for equipment capable of capacity reduction are conducted at constant environmental conditions of 80 F dry-bulb and 67 F wet-bulb, both indoor and outdoor.

Table 1: Operating Conditions for Standard Rating and Performance Tests (ARI)

TEST		INDOOR UNIT Air Entering		OUTDOOR UNIT Air Entering	
		DB °F	WB (RH) °F (%)	DB °F	WB[1](RH) °F (%)
COOLING	Standard Rating Conditions "A" Cooling, Steady State	80	67 (51%)	95	75 (40%)
	"B" Cooling, Steady State	80	67 (51%)	82	65 (40%)
	"C" Cooling, Steady State, Dry Coil	80	57 (22%)	82	65 (40%)
	"D" Cooling Cyclic, Dry Coil	80	57 (22%)	82	65 (40%)
	Maximum Operating Conditions	80	67 (51%)	115	75 (15%)
	Low Temperature Operation Cooling	67	57 (54%)	67	57 (54%)
	Part Load Conditions (IPLV)	80	67 (51%)	80	67 (51%)

[1] The wet-bulb temperature conditions is not required when testing air cooled condensers which do not evaporate condensate, or when testing without an evaporative pre-cooler.

7.3 Seasonal Energy Efficiency Ratio (SEER)

7.3.1 The SEER normally only applies to rating residential equipment rated at 65,000 Btu/h (5.4 tons) or less. However, if the tests listed above are all performed, it may be calculated for any system. This also provides the method of calculating the degradation coefficient (C_D).

7.3.2 The testing data and results required to calculate the seasonal energy efficiency ratio in Btu's per watt-hour shall include the following:

(a) Cooling capacities (Btu/h) from Tests A and B and, if applicable, the cooling capacity from Test C and the total cooling done from Test D (Btu).

(b) Electrical power input to all components and controls (watts) from Test C and the electrical usage (watt-hour) from Test D.

(c) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water).

(d) Indoor and outdoor room dry- and wet-bulb air temperatures (F)

7.3.3 Units which do not have indoor air circulating fans furnished as part of the model shall have their measured total cooling capacities adjusted by subtracting 1250 Btu/h per 1,000 CFM or measured indoor air flow and adding to the total steady-state electrical power input 365 watts per 1,000 CFM of measured indoor air flow.

7.3.4 The results of the cyclic and steady-state dry-coil performance tests shall be used in the following (4) equations:

(1) $$q_{cyc,\ dry} = \frac{60 \times Q_{mi} \times c_{pa} \times \Gamma}{[\ v'_n \times (\ 1 + W_n\ )]}$$

where the factor Γ (hr-F) is described by the equation:

(2) $$\Gamma = \int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}} [\ t_{a1} - t_{a2}\ ]_\theta\ d\theta$$

where θ represents time.

(3) $$CLF = \frac{q_{cyc,\ dry}}{q_{ss,\ dry} \times \tau}$$ Cooling Load Factor

The degradation coefficient, C_D, is then calculated as.

(4) $$C_D = \frac{1 - \dfrac{EER_{cyc,\ dry}}{EER_{ss,\ dry}}}{1 - CLF}$$

where:

$EER_{cyc,\ dry}$ = Energy efficiency ratio from Test "D", Btu/watt-hr

$EER_{ss,\ dry}$ = Energy efficiency ratio from Test "C", Btu/watt-hr

7.3.5 The SEER for units employing single-speed compressors and single-speed condenser fans shall be based on the performance of Test B and the degradation coefficient determined above. The SEER in Btu's/watt-hour shall be determined by the following equation:

$$SEER = PLF(½) \times EER_B$$

where:

EER_B = Energy efficiency ratio determined from Test B.

PLF(½) = Part-load performance factor when cooling load factor = ½ as determined from the equation:

$$PLF(½) = 1 - C_D / 2$$

7.3.6 The methodology for equipment with multiple compressors is too complex to be listed here, but the guidelines contained in the ARI Standard will be followed.

7.4 Part Load Rating (Optional)

7.4.1 Systems which are capable of capacity reduction (e.g. through cylinder unloading, multiple compressors, automatic air flow reduction) are rated at 100% and at each step of capacity reduction provided by the equipment as published by the manufacturer. These rating points are used to calculate the Integrated Part Load Value (IPLV).

7.4.2 The capacity reduction means may be adjusted to obtain the specified step of unloading. No manual adjustment of indoor and outdoor air quantities from those of the standard rating conditions shall be made. However, automatic adjustment of air quantities by system function is permissible.

7.4.3 The IPLV (in EER) shall be calculated as follows:

(a) Determine the capacity and EER at the part load conditions specified

(b) Determine the part-load factor (PLF) from Figure 2: "Part-Load Factor Curve" at each rating point.

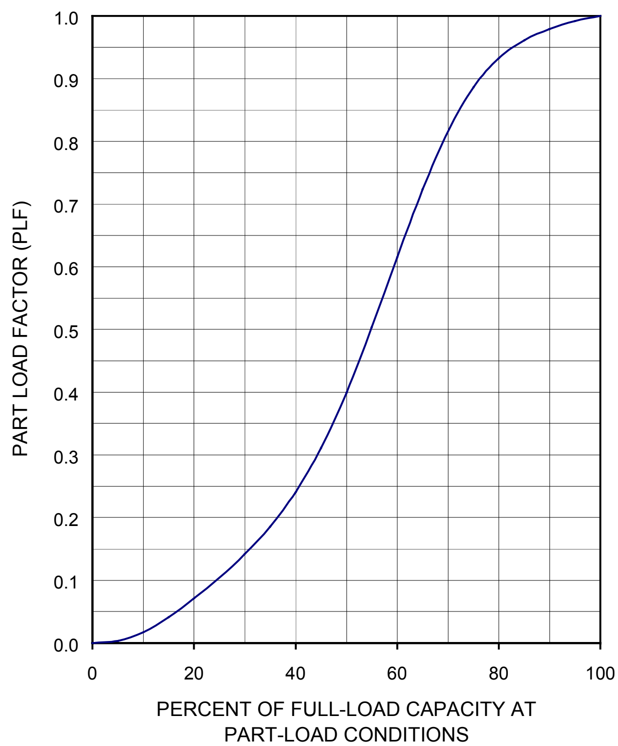
(c) Use the following equation to calculate IPLV:

$$\begin{aligned}\mathbf{IPLV} &= (\mathbf{PLF_1} - \mathbf{PLF_2})\left(\frac{\mathbf{EER_1} + \mathbf{EER_2}}{2}\right)\\ &+ (\mathbf{PLF_2} - \mathbf{PLF_3})\left(\frac{\mathbf{EER_2} + \mathbf{EER_3}}{2}\right) + \ldots\\ &+ (\mathbf{PLF_{n-1}} - \mathbf{PLF_n})\left(\frac{\mathbf{EER_{n-1}} + \mathbf{EER_n}}{2}\right)\\ &+ (\mathbf{PLF_n})(\mathbf{EER_n})\end{aligned}$$

where:

PLF	=	Part-Load Factor determined from Figure 2
n	=	Total number of capacity steps
Subscript 1	=	100% capacity and EER at part-load rating conditions
Subscript 2; 3 etc.	=	Specific capacity and EER at part-load steps.

Figure 2: Part-Load Factor Curve



7.5 Sensitivity Tests

7.5.1 Sensitivity tests expand the standard rating tests to determine the performance characteristics as a function of the environmental conditions. The adjustment factors used in DOE-2 to vary the performance characteristics from the rated conditions (Test "A") are functions of the indoor-side inlet wet-bulb temperature and the outdoor-side inlet dry-bulb temperature. Thus, the sensitivity tests are composed of a grid of test conditions which vary these parameters. The operating conditions for the sensitivity tests are listed in Table 2.

7.5.2 Variations in outdoor conditions represent changes in the outdoor environment. Variations in the indoor conditions also represent changes in the outdoor environment, as applied to different levels of economizer use. Thus, a reduction in the return air temperature can only be accomplished through mixing with cooler outdoor air. Some tests have been added to the grid for low inlet temperature operation to investigate economizer use. Since some of these are below the minimum rating conditions, it is possible that the system will not operate at these conditions. For this reason, or other testing limitations, some these low end points may not be run.

7.5.3 The performance indicators determined from this series of tests will be graphed as a function of indoor-side inlet wet-bulb temperature and outdoor-side inlet dry-bulb temperature. The data will be fit to a least-squares equation creating performance adjustment factors as a function of these variables which is applied to the performance at the standard rating conditions.

Table 2: Operating Conditions for Sensitivity Testing

		INDOOR UNIT		OUTDOOR UNIT					
		Air Entering		Air Entering					
TEST		DB °F	WB(RH) °F (%)	DB °F					
COOLING			75(80%)	115	105	95	82	67	55
		80	67(51%)	115[M]	105	95[A]	82[B]	67	55
	Steady State		57(22%)	115	105	95	82[C]	67	55
		67	57(54%)					67[L]	55
		55	46(50%)						55

Superscripts indicate ARI Standard Test Conditions (Tests A, B, C, Maximum and Low Temperature). Outdoor-side wet-bulb temperatures will correspond to the standard rating specifications for these tests, and to values which result in approximately 40% RH for the others.

7.6 Cyclic Tests

7.6.1 The cyclic testing described as Tests "C" and "D" will be expanded to include extremes of operation. Equipment literature usually states minimum limits for operating time (i.e. once switched on, it must run for some minimum time before cycling off, and similarly, once switched off, it must stay off for some minimum time before it will start again). These two minimum limits will be incorporated into a series of cyclic part load tests to describe cyclic performance as a function of the cycling frequency. The following table shows the range of cycling tests to be performed based on example values of the operating limits.

Table 3: Part Load Cycling

Limits

Minimum Off Time: 3 minutes

Minimum On Time: 5 minutes

Part Load Ratio	Time On	Time Off	Period (Min./Cycle)	Frequency (Cycles/Hour)
100%[1]	60 min.	0 min.	60	1
75%	9 min.	3 min.	12	5
50%	7½ min.	7½ min.	15	4
25%	5 min.	15 min.	20	3
20%[2]	6 min.	24 min.	30	2

[1] ARI Standard Test "C" [2] ARI Standard Cycling Test "D"

7.6.2 All of these tests are dry-coil tests; conducted at the same inlet conditions as Tests "C" and "D".

7.6.3 The results of these tests will be used to calculate a degradation coefficient for each level.

7.7 Performance Dependence on External Resistance

7.7.1 A series of tests will be conducted to determine the sensitivity of system performance to the external resistance to air flow on the indoor side. All other tests shall be conducted at the minimum external pressure listed in Table 4. The sensitivity tests will be conducted at 2, 3, and 4 times the table values, and at the environmental conditions given for standard rating Test “A”.

Table 4: Minimum External Pressure		
Standard Capacity Ratings		Minimum External Resistance
(1,000 Btu/h)	(Tons)	(Inches of Water)
0 - 28	0 - 2.3	0.10
29 - 42	2.4 - 3.5	0.15
43 - 70	3.6 - 5.8	0.20
71 - 105	5.9 - 8.8	0.25
106 - 134	8.8 - 11.2	0.30
135 - 210	11.3 - 17.5	0.35
211 - 280	17.6 - 23.3	0.40

7.8 Evaporative Pre-Cooler Effectiveness

7.8.1 If an evaporative pre-cooler is available for the test equipment, tests to determine its effectiveness will be conducted following the other tests. Since the performance of the equipment will have been determined as a function of the outdoor-side inlet dry-bulb temperature in the preceding sensitivity tests, all that remains to be determined is how well the pre-cooler works to reduce the inlet dry-bulb temperature. The overall performance of the equipment will still need to be measured to ensure that the performance still conforms to the previous results.

7.8.2 The difficulty in this test is the measurement of the air conditions downstream of the evaporative pre-cooler. The distance between the pre-cooler and the outdoor coil inlet is typically small, so it may be difficult to insert sensors or extract a representative sample. Also, effectiveness can vary considerably across the area of any evaporative cooler, so even a large array of sensors or sample locations may not accurately describe the outlet air conditions. However, the direct measurement of the outlet dry-bulb temperature may be checked against the intersection on a psychrometric chart for the inlet wet-bulb temperature, and the outlet humidity ratio as measured downstream of the outdoor coil (assuming there is no evaporation of condensate). If more accuracy is required, the cooler may be tested alone in accordance with applicable test standards (i.e. ASHRAE 133p).

7.8.3 The dry-bulb temperature of the air exiting the evaporative pre-cooler will be measured with a minimum of four temperature sensors or an equivalent sampling system. The quantity of water supplied will be measured with a totalizing liquid flow meter.

7.8.4 The test conditions will use constant indoor-side inlet conditions (80°F dry-bulb, 67°F wet-bulb), and variable levels of outdoor wet-bulb depression (difference between dry- and wet-bulb temperature), as shown in the following table.

Table 5: Outdoor-side Inlet Conditions for Evaporative Pre-Cooler Tests
[Table displays corresponding wet-bulb temperature (°F) and relative humidity]

	Wet-bulb Depression (°F)			
Dry-bulb Temperature (°F)	10	20	30	40
80	70 (61%)	60 (30%)		
95	85 (67%)	75 (40%)	65 (18%)	
115		95 (48%)	85 (30%)	75 (15%)

7.8.5 Effectiveness is calculated according to the following equation:

$$\varepsilon = \frac{t_{DB,in} - t_{DB,out}}{t_{DB,in} - t_{WB,in}}$$

8. TEST PROCEDURES

8.1 Equipment Installation

8.1.1 The equipment to be tested shall be installed in the test room or rooms in accordance with the manufacturer's installation instructions using recommended installation procedures and accessories. Air source self-contained equipment shall be located in or adjacent to an opening in the wall or partition separating the test rooms in accordance with the normal or primary recommendations of the manufacturer. The manufacturer's recommendations with respect to distances from adjacent walls, amount of extensions through walls, etc., shall be followed. All tests shall be performed at the normal voltage and frequency for which the equipment is designed.

8.1.2 No alterations to the equipment shall be made except for the attachment of required test apparatus and instruments in the prescribed manner.

8.1.3 Where necessary, equipment shall be evacuated and charged with the type and amount of refrigerant specified in the manufacturer's instructions.

8.1.4 When used, pressure gages shall be connected to the equipment only through short lengths of small diameter tubing and either shall be located so that the readings are not influenced by fluid head in the tubing or suitable cooling and heating operation corrections shall be made.

8.1.5 No change shall be made in fan speed or system resistance to correct for barometric variations.

8.2 External Resistance to Air Flow

8.2.1 The indoor-side external resistance shall be maintained at the minimum value shown in Table 4, as a function of the rated capacity of the equipment. This resistance is measured as the differential pressure between the inlet and outlet ducts to the equipment, and is maintained through adjustment of the exhaust fan and/or dampers on the air flow measurement apparatus.

8.2.2 If an outdoor-side air flow measurement apparatus is used, the outdoor-side external resistance shall be maintained within the specified tolerance at 0.0 inches of water through adjustment of the exhaust fan and/or dampers on this apparatus.

8.3 Outdoor Air for Ventilation

8.3.1 For equipment that provide for conditioning outdoor air for ventilation of the indoor space (including systems using economizers), testing should be performed with the outdoor air supply shut off. The use of outdoor air can always be simulated by adjusting the condition of the return air.

8.3.2 If shutting off this supply is not possible, such as when a separate coil is used to pre-condition the outside air, a means to measure the outdoor air flow and its properties must be provided. This may be by either direct measurement of the outdoor air flow, or through measurement of the return air flow from the indoor room, in addition to the measurement of the supply air flow.

8.3.3 If the outdoor ventilation air and the return air are thoroughly mixed before reaching the indoor coil, the proportions of the two air steams may be determined from temperature and moisture measurements, assuming suitable access to the mixed air stream is available for these measurements.

8.3.4 The indoor room shall have a means of exhausting the excess ventilation air.

8.4 Test Operating Procedure

8.4.1 The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but for not less than one hour, before capacity test data are recorded.

8.4.2 Data shall then be checked at ten minute intervals until four consecutive sets of reading within the tolerances prescribed have been obtained. The test results will be the average of all the data readings taken between the first and fourth set (i.e. 30 minutes of readings).

8.4.3 All cycling tests will be performed immediately after a steady-state test (Test C) is completed. The test unit shall be manually cycled "off" and "on" using the time periods specified until steadily repeating ambient conditions are again achieved in both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be run through an additional "off"/"on" cycle, during which the test data required shall be recorded. During this last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerances specified. During the cyclic dry-coil tests, all air moving equipment on the condenser side shall cycle "on" and "off" when the compressor cycles "on" and "off". The indoor air moving equipment shall also cycle "off" as governed by any automatic controls normally installed with the unit. This last requirement applies to units having an indoor fan time delay.

8.4.4 All electrical measurements during all "on" and "off" periods shall include auxiliary power or energy delivered to the unit (controls, transformers, crankcase heaters, etc.).

9. DATA TO BE RECORDED

9.1 Table 7 shows, generally, the data to be recorded during a test. Items indicated by an “x” under the test method columns, or their equivalent, are required when that test method is employed.

9.2 Test Tolerances

9.2.1 All test observations shall be within the tolerances specified in Table 6, as appropriate to the test methods and type of equipment.

9.2.2 The maximum permissible variation of any observation during the capacity test is listed under “Test Operating Tolerance” in the table. This represents the greatest permissible difference between maximum and minimum instrument observations during the test. When expressed as a percentage, the maximum allowable variation is the specified percentage of the arithmetical average of the observations.

9.2.3 The maximum permissible variations of the average of the test observations from the standard of desired test conditions are shown in the table under “Test Condition Tolerance”.

Table 6: Test Tolerances

Readings	**Test Condition Tolerance (Total Observed Range)**	**Test Operating Tolerance (Variation of Average From Specified Test Condition)**
Air Temperatures	°F	°F
Dry-bulb (Indoor/Outdoor, entering/leaving)[1]	2.0	0.5
Wet-bulb (Indoor/Outdoor, entering/leaving)[1,2]	1.0	0.3
Refrigerant Temperatures	3.0	0.5
Liquid Temperatures not otherwise specified	0.5	0.2
External resistance to air flow, in. Water	0.05	0.02
Electrical voltage, %	2	
Fluid flow rates, %	2	
Nozzle pressure drops, % of reading	2	

[1] For cyclic tests, operating tolerances are only applicable to inlet air temperatures after the first 30 seconds after compressor start-up.

[2] For cyclic tests, entering outdoor wet-bulb temperature operating tolerance increases to 2.0°F, and condition tolerance increases to 1.0°F.

Table 7: Data To Be Recorded

Item	Units	Indoor Air Enthalpy	Outdoor Air Enthalpy	Refrigerant Enthalpy Method	Cooling Condensate Measurement
Date/Time		x	x	x	x
Barometric Pressure	in. Hg	x	x	x	x
Power input to equipment total and components (fans, etc.)	watts or Wh	x	x	x	x
Applied voltage and frequency	volts, Hz	x	x	x	x
External resistance to air flow or coil pressure drop	in. water	x	x		x
Fan speed(s), if adjustable	rpm	x	x		x
Dry-bulb temperature of air entering equipment	°F	x	x		x
Wet-bulb temperature (or humidity) of air entering equipment	°F	x	x		x
Dry-bulb temperature of air leaving equipment	°F	x	x		x
Wet-bulb temperature (or humidity) of air leaving equipment	°F	x	x		x
Throat diameter of nozzle(s)	in.	x	x		
Velocity pressure at nozzle throat or static pressure difference across nozzle(s)	in. water	x	x		
Temperature at nozzle throat	°F	x	x		
Pressure at nozzle throat	in. Hg	x	x		
Condensing pressure or temperature	psig / °F			x	
Evaporator pressure or temperature	psig / °F			x	
Temperature of refrigerant liquid entering expansion valve	°F			x	
Temperature of refrigerant vapor entering compressor	°F			x	
Temperature of refrigerant vapor leaving compressor	°F			x	
Refrigerant-oil flow rate	ft^3/m			x	
Volume of refrigerant in refrigerant-oil mixture	ft^3 / ft^3			x	
Rate of condensate collection	lb / hr				x
Room static pressure (ref. to outside)	in. water	x	x	x	x
Evaporative pre-cooler water consumption	gph	x	x	x	x
Evaporative pre-cooler outlet dry-bulb temperature	°F	x	x	x	x

10. LETTER SYMBOLS USED IN EQUATIONS

Symbol **Description and Units**

A_n = area, nozzle, ft²
C = coefficient of discharge, nozzle
C_D = Degradation coefficient
C_{LF} = Cooling load factor.
c_{pa} = specific heat of air, Btu/(lb · °F)
E_i = power input, indoor side, watts
E_t = power input, total, watts
h_{a1} = enthalpy, air entering indoor side, Btu/lb of dry air
h_{a2} = enthalpy, air leaving indoor side, Btu/lb of dry air
h_{a3} = enthalpy, air entering outdoor side, Btu/lb of dry air
h_{a4} = enthalpy, air leaving outdoor side, Btu/lb of dry air
h_{r1} = enthalpy, refrigerant entering indoor side, Btu/lb
h_{r2} = enthalpy, refrigerant leaving indoor side, Btu/lb
P_n = pressure, at nozzle throat, in. Hg absolute
p_v = velocity pressure at nozzle throat or static pressure difference across nozzle, in. H_2O
Q_{mi} = indoor air flow rate at the dry-bulb temperature, humidity ratio, and pressure existing in the region of measurement, cfm
Q_{mo} = air flow, outdoor, measured, cfm
Q_s = air flow, standard air, cfm
$q_{cyc, dry}$ = total cooling over a cycle consisting of one compressor “off” period and one compressor “on” period (Test “D”), Btu/h
q_{lci} = latent cooling capacity, indoor side data, Btu/hr
q_{sci} = sensible cooling capacity, indoor side data, Btu/hr
$q_{ss, dry}$ = Total steady-state cooling capacity from test “C”, Btu/h
q_{tci} = total cooling capacity, indoor side data, Btu/hr
q_{tco} = total cooling capacity, outdoor side data, Btu/hr
t_{a1} = temperature, air entering indoor side, dry bulb, °F
t_{a2} = temperature, air leaving indoor side, dry bulb, °F
t_{a3} = temperature, air entering outdoor side, dry bulb, °F
t_{a4} = temperature, air leaving outdoor side, dry bulb, °F
V_r = rate of refrigerant-oil flow, ft³/hr
v_n = specific volume of air at dry and wet-bulb temperature conditions existing at nozzle but at standard barometric pressure, ft³/lb dry air
v'_n = specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate, ft³/lb
W_{i1} = humidity ratio, air entering indoor side, lb moisture per lb dry air
W_{i2} = humidity ratio, air leaving indoor side, lb moisture per lb dry air
W_n = humidity ratio, at nozzle, lb moisture per lb dry air
w_c = flow rate, indoor coil condensate, lb/hr
x = weight ratio, refrigerant to refrigerant-oil mixture
ρ = density of refrigerant, lb/ft³
θ = time, as a variable
τ = duration of time for one complete cycle consisting of one compressor “on” time and one compressor “off” time, hours

11. REFERENCE PROPERTIES AND DATA

11.1 Thermodynamic Properties of Air

The thermodynamic properties of air-water vapor mixtures shall be obtained from the latest edition of the ASHRAE Fundamentals Handbook. Alternatively, more accurate values may be obtained from the formulations of Hyland and Wexler (1983), or from a computer program derived from these formulations.

11.2 Thermodynamic Properties of Water and Steam

The thermodynamic properties of water and steam shall be obtained from the latest edition of the ASHRAE Fundamentals Handbook, the ASME Steam Tables, or a computer program derived from these sources.

11.3 Thermodynamic Properties of Volatile Refrigerants

The thermodynamic properties of volatile refrigerants shall be obtained from the latest edition of the ASHRAE Fundamentals Handbook, or the ASHRAE Thermodynamic Properties of Refrigerants.

12. REFERENCES

ANSI/ASHRAE 37-1988, *"Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1988.

ANSI/ASHRAE 41.1-1986, *"Standard Method for Temperature Measurement"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1986.

ANSI/ASHRAE 41.2-1987, *"Standard Methods for Laboratory Airflow Measurement"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1987.

ANSI/ASHRAE 41.6-1994, *"Method for Measurement of Moist Air Properties"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1994.

ANSI/AMCA 210-85 | ANSI/ASHRAE 51-1985, *"Laboratory Methods of Testing Fans for Rating"*, Jointly published by: Air Movement and Control Association, Inc., 30 West University Drive, Arlington Heights, IL 60004, and: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1985.

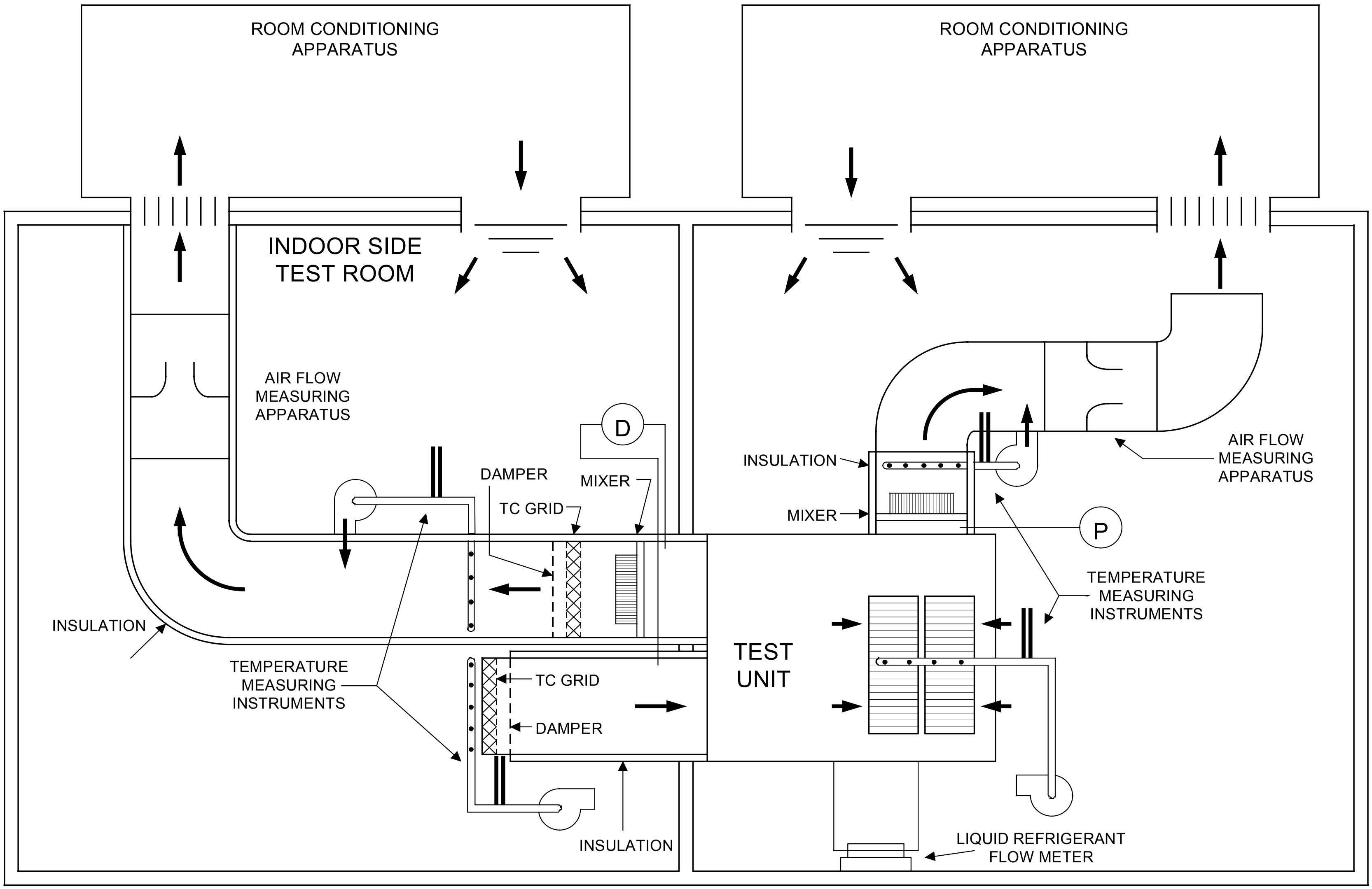
ANSI/ASHRAE 116-1983, *"Methods of Testing for Seasonal Efficiency of Unitary Air-Conditioners and Heat Pumps"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1983.

ARI Standard 210/240-94, *"Unitary Air-Conditioning and Air-Source Heat Pump Equipment"*, Air-Conditioning and Refrigeration Institute, 4301 North Fairfax Drive, Arlington, VA 22203, 1994.

ARI Standard 340/360-93, *"Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment"*, Air-Conditioning and Refrigeration Institute, 4301 North Fairfax Drive, Arlington, VA 22203, 1993.

10 CFR 11, Part 430, Subpart B, Appendix M *"Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners"* Department of Energy, 1998.

Attachment 1: Arrangement for the Air-enthalpy Test Method



INSTRUMENTATION INFORMATION DATA SHEET

Test Description: ____

Test Instrument	Model # & Inventory # / Serial #	Calibration Due Date
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	
	Model No.: ____________ Inv./Serial No.: ____________ Range: ____________ Accuracy: ____________	

PRESSURE CALIBRATION DATA SHEET

Test Description: ______________________________

Instrument Description: ______________________________

Initial Calibration Data:

Date: ______________ Time: ______________ Taken By: ______________

	Standard	**Transmitter Reading**
Low Limit Pressure		
High Limit Pressure		

Pre-test Calibration Data:

Date: ______________ Time: ______________ Taken By: ______________

	Standard	**Transmitter Reading**	**Data Logger Reading**
Low Limit Pressure			
25% of Range Pressure			
50% of Range Pressure			
75% of Range Pressure			
High Limit Pressure			

Post-Test Calibration Data:

Date: ______________ Time: ______________ Taken By: ______________

	Standard	**Transmitter Reading**	**Data Logger Reading**
Low Limit Pressure			
High Limit Pressure			

Instrument Calibration Standards:

Description	TES Inventory No. or Serial No.

Comments:

TEMPERATURE CALIBRATION DATA SHEET

Test Description: ______________________________

Instrument Description: ______________________________

Initial Calibration Data:

Date: ________________ Time: ________________ Taken By: ________________

	Standard	**Reading**
Ice Point		
High Point Temperature		

Pre-test Calibration Data:

Date: ________________ Time: ________________ Taken By: ________________

	Standard	**Reading**	**Difference**
Ice Point			
High Point Temperature			

Post-Test Calibration Data:

Date: ________________ Time: ________________ Taken By: ________________

	Standard	**Reading**	**Difference**
Ice Point			
High Point Temperature			

Instrument Calibration Standards:

Description	TES Inventory No. or Serial No.

Comments:

APPENDIX III

Measurement System Description

Data Acquisition System

Manufacturer	Equipment	Model
National Instruments	Data Acquisition Card	PCI-MIO-16XE-50
	Signal Conditioning	SCXI System - 1001 Chassis 1102 32-Channel Thermocouple Amplifier Module (2) 1124 6-Channel Analog Output Module (2)
	GPIB Card	AT-GPIB/TNT
	DAQ Programming Environment Software	LabVIEW 5.0
Hewlett Packard	Data Logger	34970A (2)
	Input Cards	34901A 20+2 Channel Input Module (3) 34902A 16-Channel high-speed Input Module (2) 34907A Multifunction Module
Gateway 2000	Personal Computer	P5-120 64 MB Ram, 2 GB Disk OS: Microsoft Windows 95

Instrumentation

Measurements	Device Type	Make/Model
Evaporator Inlet Dry Bulb Temperatures (4) Evaporator Outlet Dry Bulb Temperatures (4) Evaporator Inlet and Outlet Sample Dry Bulb Temperatures Condenser Inlet Dry Bulb Temperatures (4) Evaporator and Condenser Airflow Chamber Inlet Temperatures Indoor and Outdoor Room Conditioning Systems Supply and Return Duct Temperatures	Resistance Temperature Detectors (¼-inch RTD probes with fast response tips)	Burns Engineering WSP0G21-5D
Evaporator Inlet and Outlet Sample Wet Bulb Temperatures Condenser Inlet Wet Bulb Temperatures (4) Condenser Outlet Sample Dry and Wet Bulb Temperature Ambient Dry and Wet Bulb Temperatures	Resistance Temperature Detectors (¼-inch RTD probes)	Rosemount 78S01N0900 & 78N01N00N090

Instrumentation (Continued)

Measurements	**Device Type**	**Make/Model**
Evaporator Inlet Dry Bulb Temperature (9) Evaporator Outlet Dry Bulb Temperature (9) Condenser Inlet Dry Bulb Temperature (after pre-cooler; 3 sensors with 3 junctions in parallel) Condenser Outlet Dry Bulb (4 sensors with 3 junctions in parallel) Evaporator Coil Surface Temperature (2)	Type "T" Thermocouple	Thermax
Evaporator Inlet Sample Dew Point Temperature Evaporator Outlet Sample Dew Point Temperature	Refrigerated Mirror Hygrometer	General Eastern Hygro-M1 (in) & Hygro-M2 (out)
Evaporator Outlet Sample Relative Humidity Condenser Outlet Sample Relative Humidity	Capacitive Relative Humidity Sensor	Vaisala HMP-233
Condenser Inlet Relative Humidity	Resistive Relative Humidity Sensor	General Eastern MRH-2-OA
Evaporator and Condenser Outlet Airflow Chamber Static and Differential Pressure Evaporator and Condenser External Resistance	"Smart" Pressure Transmitters	Rosemount 3051 - CD1A22A1AE5
Barometric Pressure (Also Ambient Temperature, Relative Humidity, and Wind Speed and Direction)	Packaged Weather Station	Qualimetrics
Total Real Power Total Reactive Power Line-to-Line Voltages Line Currents Line Frequency	3-Phase Digital Power Meter	Yokogawa 2533
Compressor(s) Power Evaporator Fan Power Condenser Fan Power	Watt/Watt-hour Transducers	Scientific Columbus - Digilogic DL342K5A26070V
Pre-cooler Water Use	Positive Displacement Flow Meter (with pulse output)	Kent 5/8 × 3/4"

Field Calibration Instruments

Measurements	Devices
Temperature	Fluke 2180A RTD Temperature Standard Rosemount Temperature Calibration Bath Gallium Melting Point Cell (85.6°F) Ice (32°F)
Pressure	PPC 500 Pressure Calibrator Dwyer Microtector

Electrical measurement instruments were calibrated
by the TES Standards Lab.

Dew point sensors were returned to the manufacturer for calibration.

Relative humidity sensors used as-delivered factory calibrations.

APPENDIX IV

Test Data

Test Unit #1 - Baseline Unit

Test Summary Information	Low Temperature Tests					Low Evaporator Inlet Humidity ("Dry" Coil) Tests													
Nominal Test Conditions					"Low"														
Condenser Inlet Dry Bulb (°F)	55	55	55	55	67	55	55	67	67	82	82	82	95	95	105	105	105	115	115
Evaporator Inlet Dry Bulb (°F)	55	55	67	67	67	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	46	46	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General																			
Date	6/9/99	6/14/99	6/9/99	6/13/99	6/13/99	6/9/99	6/13/99	6/9/99	6/13/99	6/9/99	6/12/99	6/16/99	6/10/99	6/12/99	6/10/99	6/12/99	6/16/99	6/10/99	6/12/99
Start Time	9:29a	10:48a	12:00p	11:25a	1:30p	2:41p	10:21a	4:29p	9:04a	5:38p	10:15a	8:14a	2:56p	3:27p	4:10p	4:24p	6:14p	5:16p	5:14p
Duration (minutes)	30	21	30	30	30	35	30	30	30	31	30	30	30	30	30	30	30	33	30
Barometric Pressure (in. of Hg)	29.56	29.59	29.55	29.66	29.64	29.54	29.66	29.53	29.67	29.53	29.65	29.55	29.54	29.64	29.52	29.66	29.49	29.52	29.66
Condenser Air Properties																			
Inlet Dry Bulb Temperature (°F)	55.2	55.0	55.0	55.0	67.0	55.0	55.0	66.9	67.7	82.0	82.0	82.0	95.0	95.0	105.0	105.0	104.9	115.0	115.1
Inlet Relative Humidity (%)	69%	70%	63%	76%	57%	62%	72%	46%	53%	31%	35%	36%	23%	24%	14%	18%	18%	3%	11%
Outlet Dry Bulb Temperature (°F)	72.5	72.2	74.5	74.8	87.5	75.6	75.5	88.5	89.2	103.9	104.0	104.0	117.4	117.3	126.9	127.1	127.0	136.6	136.9
Air Flow Rate (CFM)	5,280	5,270	5,290	5,270	5,270	5,290	5,270	5,290	5,250	5,300	5,290	5,270	5,270	5,250	5,240	5,240	5,260	5,230	5,240
Evaporator Air Properties																			
Inlet Dry Bulb Temperature (°F)	55.0	55.0	67.0	67.1	67.4	80.0	80.0	80.0	80.0	80.0	80.1	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	34.4	37.4	43.7	49.1	49.3	43.6	46.6	44.5	47.3	46.1	48.9	50.4	48.5	48.2	46.1	50.3	52.2	45.5	51.5
Inlet Wet Bulb Temperature (°F) - calc.	45.2	46.3	54.1	56.6	56.8	59.1	60.3	59.5	60.6	60.1	61.3	61.9	61.1	61.0	60.1	61.9	62.7	59.8	62.5
Outlet Dry Bulb Temperature (°F)	37.4	38.0	46.5	48.7	49.3	54.8	55.5	55.3	55.7	56.9	56.9	57.0	58.9	58.9	60.4	60.4	61.0	62.2	62.1
Outlet Dew Point Temperature (°F)	32.4	33.8	40.7	44.3	44.9	43.2	45.7	44.3	46.6	46.0	48.4	49.3	48.3	48.3	46.0	50.1	51.8	45.4	51.5
Air Flow Rate (CFM)	3,430	3,360	3,360	3,300	3,330	3,500	3,470	3,540	3,480	3,550	3,460	3,370	3,560	3,560	3,560	3,560	3,580	3,560	3,560
Power Consumption																			
Total Demand (kW)	8.16	8.12	8.34	8.39	9.13	8.55	8.55	9.35	9.40	10.26	10.21	10.10	11.02	11.02	11.56	11.57	11.53	12.09	12.10
Compressor (kW)	5.63	5.66	5.88	5.99	6.72	6.04	6.09	6.83	6.94	7.74	7.77	7.72	8.51	8.51	9.07	9.07	9.06	9.61	9.62
Evaporator Blower (kW)	1.96	1.90	1.89	1.83	1.85	1.93	1.89	1.95	1.90	1.96	1.88	1.82	1.95	1.94	1.94	1.94	1.91	1.93	1.93
Condenser Fan (kW)	0.64	0.65	0.65	0.65	0.65	0.64	0.64	0.63	0.63	0.62	0.63	0.65	0.63	0.63	0.62	0.62	0.63	0.62	0.62
Performance																			
Total Cooling Capacity (Tons)	6.23	6.32	7.33	7.36	7.23	8.22	8.13	8.06	8.03	7.53	7.51	7.51	6.91	6.86	6.36	6.38	6.29	5.77	5.80
Sensible Cooling Factor	92%	86%	87%	77%	77%	99%	96%	99%	97%	100%	97%	94%	99%	100%	99%	99%	98%	99%	100%
Coil Bypass Factor	24%	24%	25%	25%	25%	32%	29%	31%	28%	32%	27%	26%	33%		43%	35%	33%	49%	37%
Energy Efficiency Ratio (Btu/Wh)	9.16	9.33	10.54	10.53	9.50	11.54	11.41	10.35	10.25	8.81	8.83	8.92	7.52	7.47	6.60	6.62	6.55	5.73	5.75

Test Unit #1 - Baseline Unit

Test Summary Information	Moderate Evaporator Inlet Humidity Tests												
Nominal Test Conditions		------ "B" ------			----------------- "A" -----------------						------ "Max" ------		
Condenser Inlet Dry Bulb (°F)	55	67	67	82	82	95	95	95	95	105	105	115	115
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67	67	67
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General													
Date	6/11/99	6/11/99	7/23/99	6/11/99	7/23/99	6/11/99	6/15/99	6/15/99	7/23/99	6/11/99	7/23/99	6/11/99	7/23/99
Start Time	4:55p	3:54p	9:20a	2:35p	12:31p	10:56a	9:34a	3:07p	1:56p	11:59a	3:12p	12:57p	4:52p
Duration (minutes)	30	31	30	30	30	30	30	30	30	30	30	30	30
Barometric Pressure (in. of Hg)	29.57	29.57	29.72	29.57	29.70	29.59	29.55	29.53	29.70	29.58	29.68	29.58	29.69
Condenser Air Properties													
Inlet Dry Bulb Temperature (°F)	55.0	67.0	67.0	82.0	82.3	95.1	95.0	95.0	95.0	105.0	105.0	115.0	114.9
Inlet Relative Humidity (%)	70%	54%	49%	31%	33%	24%	25%	23%	24%	16%	16%	7%	5%
Outlet Dry Bulb Temperature (°F)	76.0	89.2	89.9	105.0	105.8	118.1	118.2	118.1	118.4	127.6	128.0	137.0	137.5
Air Flow Rate (CFM)	5,240	5,240	5,060	5,220	5,060	5,250	5,260	5,210	5,060	5,230	5,060	5,230	5,020
Evaporator Air Properties													
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.1	60.2	59.9	60.2	60.0	60.1	60.2	60.2	60.2	60.1	60.5	60.1	60.7
Inlet Wet Bulb Temperature (°F) - calc.	66.8	66.9	66.7	66.9	66.8	66.8	66.9	66.9	66.9	66.8	67.1	66.8	67.2
Outlet Dry Bulb Temperature (°F)	60.3	60.5	60.1	61.2	60.7	62.1	62.0	61.8	61.9	62.9	62.8	63.9	63.8
Outlet Dew Point Temperature (°F)	55.7	55.6	55.1	55.9	55.5	56.5	56.4	56.3	56.3	57.0	57.2	57.7	57.8
Air Flow Rate (CFM)	3,340	3,330	3,370	3,340	3,310	3,360	3,300	3,260	3,360	3,350	3,350	3,360	3,370
Power Consumption													
Total Demand (kW)	8.54	9.40	9.44	10.37	10.38	11.13	11.04	11.03	11.12	11.64	11.64	12.13	12.15
Compressor (kW)	6.13	7.00	7.03	7.98	8.01	8.73	8.69	8.69	8.73	9.25	9.27	9.75	9.78
Evaporator Blower (kW)	1.82	1.81	1.81	1.82	1.79	1.83	1.78	1.77	1.81	1.82	1.80	1.82	1.80
Condenser Fan (kW)	0.64	0.64	0.67	0.63	0.66	0.63	0.66	0.65	0.66	0.62	0.66	0.63	0.65
Performance													
Total Cooling Capacity (Tons)	8.25	8.25	8.60	7.93	8.13	7.36	7.39	7.33	7.53	6.80	7.01	6.18	6.49
Sensible Cooling Factor	72%	71%	71%	72%	72%	74%	73%	73%	73%	76%	75%	79%	76%
Coil Bypass Factor	24%	25%	26%	28%	27%	28%	29%	28%	28%	30%	29%	31%	31%
Energy Efficiency Ratio (Btu/Wh)	11.58	10.53	10.93	9.17	9.40	7.94	8.03	7.98	8.13	7.01	7.22	6.11	6.41

"B" Avg. Capacity	8.03	"A" Avg. Capacity	7.40	"Max" Avg. Capacity	6.33
"B" Avg. Demand	10.4	"A" Avg. Demand	11.1	"Max" Avg. Demand	12.1
"B" Avg. EER	9.28	"A" Avg. EER	8.02	"Max" Avg. EER	6.26
"B" Avg. Flow	3,330	"A" Avg. Flow	3,320	"Max" Avg. Flow	3,360

Test Unit #1 - Baseline Unit

Test Summary Information	High Evaporator Inlet Humidity Tests											
Nominal Test Conditions												
Condenser Inlet Dry Bulb (°F)	55	67	67	82	82	95	95	95	105	105	115	115
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	75	75	75	75	75	75	75	75	75	75	75	75
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General												
Date	6/13/99	6/13/99	7/26/99	6/13/99	7/26/99	6/14/99	7/26/99	7/26/99	6/14/99	7/26/99	6/14/99	7/26/99
Start Time	5:26p	4:26p	10:35a	6:28p	11:39a	1:10p	12:43p	1:22p	2:19p	2:27p	3:36p	3:55p
Duration (minutes)	30	30	30	30	31	30	38	34	30	30	30	32
Barometric Pressure (in. of Hg)	29.60	29.61	29.57	29.60	29.56	29.57	29.56	29.55	29.56	29.54	29.54	29.53
Condenser Air Properties												
Inlet Dry Bulb Temperature (°F)	55.0	67.0	67.0	82.1	82.1	95.0	95.1	95.7	105.0	105.0	115.0	115.1
Inlet Relative Humidity (%)	81%	58%	53%	46%	38%	28%	27%	26%	21%	19%	12%	9%
Outlet Dry Bulb Temperature (°F)	76.4	90.0	90.6	106.4	106.8	119.9	120.3	120.9	129.9	130.2	139.6	140.2
Air Flow Rate (CFM)	5,280	5,270	5,110	5,270	5,090	5,240	5,070	5,070	5,230	5,060	5,210	5,010
Evaporator Air Properties												
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	72.9	73.0	73.0	72.9	73.1	73.0	73.0	73.0	72.9	72.9	72.8	72.8
Inlet Wet Bulb Temperature (°F) - calc.	74.9	74.9	74.9	74.8	75.0	74.9	74.9	74.9	74.8	74.9	74.8	74.8
Outlet Dry Bulb Temperature (°F)	67.8	68.0	67.8	68.3	68.4	69.2	69.1	69.2	69.9	69.8	70.7	70.5
Outlet Dew Point Temperature (°F)	65.8	65.7	65.2	65.6	65.5	65.8	65.8	65.9	66.1	66.1	66.5	66.5
Air Flow Rate (CFM)	3,320	3,310	3,290	3,280	3,250	3,240	3,250	3,270	3,240	3,280	3,250	3,290
Power Consumption												
Total Demand (kW)	8.57	9.50	9.53	10.57	10.59	11.35	11.43	11.48	11.98	12.06	12.59	12.67
Compressor (kW)	6.23	7.17	7.26	8.28	8.30	9.09	9.15	9.22	9.73	9.78	10.34	10.38
Evaporator Blower (kW)	1.77	1.76	1.72	1.74	1.72	1.72	1.71	1.71	1.72	1.72	1.72	1.73
Condenser Fan (kW)	0.69	0.68	0.68	0.67	0.67	0.65	0.66	0.66	0.65	0.66	0.57	0.65
Performance												
Total Cooling Capacity (Tons)	8.83	8.84	9.12	8.60	8.79	8.18	8.32	8.30	7.73	7.90	7.17	7.33
Sensible Cooling Factor	41%	40%	40%	40%	39%	38%	38%	38%	38%	38%	37%	38%
Coil Bypass Factor	38%	44%										
Energy Efficiency Ratio (Btu/Wh)	12.36	11.16	11.48	9.76	9.96	8.64	8.73	8.68	7.74	7.87	6.83	6.94

Test Unit #1 - Baseline Unit

Test Summary Information	Variable Evaporator Resistance					Cycling Tests				
Nominal Test Conditions						"C"				"D"
Condenser Inlet Dry Bulb (°F)	95	95	95	95	95	82	82	82	82	82
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	57	57	57	57	57
Evaporator External Resistance (IW)	0.25	0.50	0.75	1.00	0.25	0.25	0.25	0.25	0.25	0.25
General										
Date	6/15/99	6/15/99	6/15/99	6/15/99	6/15/99	6/12/99	6/12/99	6/12/99	6/12/99	6/12/99
Start Time	9:34a	10:15a	10:55a	11:47a	3:07p	10:15a	10:46a	11:25a	12:33p	1:46p
Duration (minutes)	30	14	30	30	30	30	37	58	70	78
Barometric Pressure (in. of Hg)	29.55	29.55	29.55	29.54	29.53	29.65	29.66	29.66	29.66	29.65
Condenser Air Properties										
Inlet Dry Bulb Temperature (°F)	95.0	95.2	95.3	95.1	95.0	82.0	82.3	82.5	82.6	82.8
Inlet Relative Humidity (%)	25%	25%	25%	25%	23%	35%	35%	35%	34%	32%
Outlet Dry Bulb Temperature (°F)	118.2	118.1	117.6	116.9	118.1	104.0	103.1	102.5	101.3	101.8
Air Flow Rate (CFM)	5,260	5,240	5,230	5,220	5,210	5,290	5,250	5,250	5,230	5,240
Evaporator Air Properties										
Inlet Dry Bulb Temperature (°F)	80.0	80.1	80.3	80.0	80.0	80.1	80.0	79.9	80.0	79.8
Inlet Dew Point Temperature (°F)	60.2	60.2	60.1	60.0	60.2	48.9	49.6	50.0	49.0	47.7
Inlet Wet Bulb Temperature (°F) - calc.	66.9	66.9	66.9	66.8	66.9	61.3	61.6	61.7	61.3	60.7
Outlet Dry Bulb Temperature (°F)	62.0	61.0	59.8	58.0	61.8	56.9	58.9	59.9	61.5	60.6
Outlet Dew Point Temperature (°F)	56.4	55.5	54.5	52.9	56.3	48.4				
Air Flow Rate (CFM)	3,300	2,950	2,590	2,200	3,260	3,460	3,510	3,540	3,540	3,550
Power Consumption							↓ kWh / hr ↓			
Total Demand (kW)	11.04	10.85	10.63	10.34	11.03	10.21	7.83	5.27	2.74	2.20
Compressor (kW)	8.69	8.65	8.59	8.47	8.69	7.77				
Evaporator Blower (kW)	1.78	1.63	1.47	1.31	1.77	1.88				
Condenser Fan (kW)	0.66	0.65	0.64	0.65	0.65	0.63				
Performance							↓ Ton-hr / hr (Sensible) ↓			
Total Cooling Capacity (Tons)	7.39	7.26	7.06	6.75	7.33	7.32	5.47	3.51	1.69	1.37
Sensible Cooling Factor	73%	70%	68%	65%	73%					
Coil Bypass Factor	29%	29%	27%	27%	28%					
Energy Efficiency Ratio (Btu/Wh)	8.03	8.03	7.97	7.83	7.98	8.61	8.39	8.00	7.42	7.49
			Cycle Period:			30	12	15	20	30
			"On" Fraction:			100%	75%	50%	25%	20%
			"On" Time (minutes):			30	9	7.5	5	6
			Cooling Load Factor:				74.8%	48.0%	23.1%	18.8%
			Degradation Coefficient:				0.099	0.135	0.180	0.159
			SEER:							8.54

Test Unit #2 - Baseline Unit with Evaporative Pre-cooler

Test Summary Information	82°F Outside Dry Bulb Tests										95°F Outside Dry Bulb Tests											
Nominal Test Conditions	"B" (interpolated)										"A"											
Condenser Inlet Dry Bulb (°F)	82	82	82	82	82	82	82	82	82	82	95	95	95	95	95	95	95	95	95	95	95	95
Condenser Inlet Wet Bulb (°F)	60	60	60	60	70	70	70	70	70	70	65	65	65	65	75	75	75	85	85	85	85	85
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General																						
Date	6/25/99	7/2/99	7/7/99	7/16/99	6/24/99	6/24/99	6/24/99	6/25/99	7/2/99	7/6/99	6/29/99	7/7/99	7/16/99	7/20/99	6/25/99	7/6/99	7/20/99	6/28/99	6/28/99	7/6/99	7/15/99	7/15/99
Start Time	5:00p	9:46a	6:17p	1:38p	2:05p	3:32p	5:34p	9:14a	3:18p	12:57p	9:46a	12:59p	3:13p	10:05a	10:57a	2:42p	3:31p	1:39p	2:23p	4:08p	11:16a	12:46p
Duration (minutes)	31	30	20	35	84	33	48	31	14	41	40	31	35	35	32	32	30	13	31	24	16	30
Barometric Pressure (in. of Hg)	29.46	29.34	29.49	29.57	29.48	29.47	29.47	29.51	29.34	29.56	29.51	29.57	29.55	29.62	29.51	29.55	29.59	29.51	29.50	29.54	29.51	29.51
Condenser Air Properties																						
Inlet Dry Bulb Temperature (°F)	81.9	82.0	82.0	82.0	82.0	82.0	82.0	82.0	81.8	82.1	95.0	95.1	95.0	95.0	95.0	95.0	95.0	94.8	95.1	95.0	94.9	95.0
Inlet Wet Bulb Temperature (°F)	60.7	64.2	61.6	64.4	67.9	68.0	68.3	68.6	69.8	70.1	68.0	65.5	68.3	69.3	75.1	76.6	75.1	84.9	85.1	85.2	85.1	85.3
Inlet Relative Humidity (%)	28%	39%	31%	39%	50%	50%	51%	52%	56%	57%	25%	20%	26%	28%	41%	45%	41%	67%	67%	68%	68%	68%
Outlet Dry Bulb Temperature (°F)	96.3	99.1	97.7	98.0	99.9	100.0	100.1	100.3	100.9	101.3	107.1	105.9	106.9	107.6	110.3	111.0	110.3	115.0	115.1	115.2	115.2	115.4
Air Flow Rate (CFM)	4,820	4,840	4,770	4,760	4,870	4,890	4,870	4,870	4,800	4,810	4,800	4,750	4,770	4,780	4,850	4,790	4,710	4,810	4,800	4,780	4,770	4,760
Evaporator Air Properties																						
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.1	61.4	65.6	60.1	60.1	60.2	60.2	60.1	58.8	60.4	60.2	60.2	60.2	60.1	60.1	60.3	60.2	60.2	60.0	60.4	60.5	60.3
Inlet Wet Bulb Temperature (°F) - calc.	66.8	67.5	70.0	66.8	66.8	66.9	66.9	66.8	66.1	67.0	66.8	66.9	66.9	66.9	66.8	66.9	66.9	66.9	66.8	67.0	67.0	66.9
Outlet Dry Bulb Temperature (°F)	60.3	61.2	63.6	60.6	60.5	60.6	60.6	60.5	59.9	60.9	61.0	61.0	61.2	61.2	61.1	61.3	61.4	61.8	61.5	61.8	61.8	61.7
Outlet Dew Point Temperature (°F)	55.1	56.3	59.3	55.4	55.6	55.7	55.7	55.6	54.4	55.9	55.5	55.6	55.7	55.8	56.0	56.3	56.3	57.1	56.8	57.0	57.3	57.0
Air Flow Rate (CFM)	3,320	3,350	3,310	3,360	3,290	3,290	3,300	3,300	3,330	3,310	3,300	3,300	3,360	3,350	3,300	3,320	3,340	3,280	3,290	3,320	3,330	3,320
Power Consumption																						
Total Demand (kW)	9.90	10.08	10.04	10.06	10.10	10.11	10.12	10.11	10.15	10.24	10.54	10.47	10.58	10.58	10.70	10.81	10.74	10.97	10.97	11.05	11.03	11.03
Compressor (kW)	7.41	7.59	7.57	7.55	7.64	7.64	7.64	7.65	7.67	7.77	8.05	7.99	8.07	8.10	8.24	8.34	8.25	8.51	8.51	8.58	8.54	8.61
Evaporator Blower (kW)	1.77	1.79	1.78	1.81	1.76	1.77	1.77	1.76	1.78	1.78	1.78	1.79	1.81	1.79	1.77	1.79	1.79	1.76	1.76	1.79	1.79	1.77
Condenser Fan (kW)	0.66	0.68	0.68	0.68	0.66	0.67	0.68	0.68	0.67	0.68	0.68	0.68	0.64	0.66	0.67	0.67	0.67	0.67	0.67	0.66	0.67	0.66
Unit Performance																						
Total Cooling Capacity (Tons)	8.48	8.32	8.56	8.33	8.09	8.08	8.10	8.13	8.18	8.09	8.05	8.01	8.03	7.93	7.75	7.66	7.65	7.03	7.15	7.27	7.13	7.18
Sensible Cooling Factor	69%	68%	57%	71%	72%	72%	71%	72%	74%	71%	71%	71%	71%	72%	73%	73%	73%	77%	77%	75%	77%	76%
Coil Bypass Factor	28%	27%	32%	27%	25%	25%	26%	25%	26%	26%	28%	28%	28%	28%	26%	26%	26%	24%	24%	25%	24%	24%
Energy Efficiency Ratio (Btu/Wh)	10.3	9.9	10.2	9.9	9.6	9.6	9.6	9.7	9.7	9.5	9.2	9.2	9.1	9.0	8.7	8.5	8.6	7.7	7.8	7.9	7.8	7.8
Precooler Performance																						
Outlet Dry Bulb Temperature (°F)	71.4	73.6	72.3	72.4	75.1	75.3	75.4	75.3	76.0	76.6	82.1	80.7	80.8	81.4	84.8	85.9	84.5	90.2	90.4	90.3	89.5	89.7
Temperature Drop (°F)	10.5	8.5	9.6	9.6	6.9	6.7	6.6	6.7	5.8	5.5	12.9	14.4	14.2	13.6	10.2	9.1	10.5	4.7	4.7	4.6	5.4	5.3
Effectiveness	49%	47%	47%	54%	49%	48%	48%	50%	48%	46%	48%	49%	53%	53%	52%	49%	53%	47%	47%	48%	55%	55%
Water Use Rate (GPH)	9.8	8.3	9.9	8.5	7.6	7.6	7.7	7.3	6.8	6.7	10.9	11.6	10.8	11.4	9.1	8.5	8.9	6.6	6.7	6.4	6.5	6.4

"B" Avg. Capacity: 8.30
"B" Avg. Power: 10.1
"B" Avg. EER: 9.90
"B" Avg. Evap. Flow: 3,320
"B" Avg. Cond. Flow: 4,820

"A" Avg. Capacity: 7.69
"A" Avg. Power: 10.7
"A" Avg. EER: 8.58
"A" Avg. Evap. Flow: 3,320
"A" Avg. Cond. Flow: 4,780

Test Unit #2 - Baseline Unit with Evaporative Pre-cooler

Test Summary Information	105°F Outside Dry Bulb Tests										115°F Outside Dry Bulb Tests									
Nominal Test Conditions											---- "Max" ----									
Condenser Inlet Dry Bulb (°F)	105	105	105	105	105	105	105	105	105	105	115	115	115	115	115	115	115	115	115	115
Condenser Inlet Wet Bulb (°F)	70	70	70	80	80	80	90	90	90	90	75	75	85	85	85	85	95	95	95	95
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General																				
Date	6/29/99	7/7/99	7/20/99	6/25/99	7/6/99	7/15/99	6/28/99	7/6/99	7/15/99	7/16/99	7/7/99	7/20/99	6/25/99	6/29/99	7/7/99	7/16/99	6/29/99	7/7/99	7/16/99	7/16/99
Start Time	11:24a	2:07p	12:28p	1:52p	5:18p	2:51p	4:37p	7:06p	4:43p	5:28p	3:28p	2:01p	3:15p	1:56p	9:15a	8:40a	3:53p	10:55a	10:16a	10:47a
Duration (minutes)	31	31	30	32	30	31	21	46	30	32	45	30	31	30	35	31	20	34	19	31
Barometric Pressure (in. of Hg)	29.50	29.55	29.62	29.49	29.54	29.49	29.48	29.55	29.47	29.54	29.53	29.61	29.47	29.49	29.60	29.57	29.47	29.59	29.58	29.58
Condenser Air Properties																				
Inlet Dry Bulb Temperature (°F)	105.1	105.0	105.0	105.1	105.0	105.0	105.0	104.9	105.0	105.0	115.0	114.9	115.0	115.0	115.1	115.0	115.0	115.2	115.2	115.3
Inlet Wet Bulb Temperature (°F)	72.3	67.6	73.2	79.6	79.5	77.2	90.1	86.2	90.0	89.8	70.5	78.5	84.1	85.6	84.9	84.7	94.9	91.3	95.1	95.0
Inlet Relative Humidity (%)	20%	12%	23%	34%	34%	29%	57%	48%	57%	56%	6%	20%	29%	31%	30%	30%	48%	41%	49%	48%
Outlet Dry Bulb Temperature (°F)	114.2	111.8	114.3	117.1	117.3	116.1	122.5	120.5	122.3	122.2	118.2	121.5	124.1	125.1	124.5	124.2	129.5	127.6	129.2	129.2
Air Flow Rate (CFM)	4,790	4,740	4,730	4,810	4,760	4,730	4,770	4,730	4,700	4,730	4,740	4,710	4,780	4,740	4,730	4,770	4,730	4,710	4,720	4,710
Evaporator Air Properties																				
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.2	60.0	60.0	60.1	60.6	59.9	60.1	60.2	60.2	60.3	60.3	60.2	60.0	60.1	59.7	60.1	60.0	59.6	60.4	60.4
Inlet Wet Bulb Temperature (°F) - calc.	66.9	66.8	66.8	66.8	67.1	66.7	66.8	66.9	66.8	66.9	66.9	66.9	66.8	66.8	66.6	66.8	66.8	66.6	67.0	67.0
Outlet Dry Bulb Temperature (°F)	61.4	61.3	61.6	61.8	62.0	61.7	62.4	62.1	62.5	62.7	62.0	62.6	62.5	62.7	62.4	62.7	63.3	62.6	63.4	63.4
Outlet Dew Point Temperature (°F)	55.9	55.7	56.1	56.5	56.9	56.3	57.6	57.1	57.5	57.7	56.2	56.9	57.0	57.3	57.0	57.2	58.2	57.4	58.4	58.3
Air Flow Rate (CFM)	3,280	3,320	3,320	3,320	3,330	3,350	3,300	3,320	3,340	3,360	3,330	3,360	3,330	3,340	3,340	3,360	3,340	3,330	3,340	3,340
Power Consumption																				
Total Demand (kW)	10.93	10.81	10.94	11.09	11.17	11.10	11.37	11.32	11.42	11.43	11.16	11.36	11.46	11.54	11.48	11.50	11.77	11.66	11.78	11.79
Compressor (kW)	8.45	8.35	8.46	8.61	8.69	8.60	8.90	8.86	8.95	8.94	8.69	8.87	8.98	9.05	9.02	9.02	9.29	9.20	9.32	9.32
Evaporator Blower (kW)	1.76	1.79	1.78	1.78	1.80	1.81	1.77	1.78	1.80	1.80	1.80	1.80	1.78	1.79	1.80	1.80	1.79	1.80	1.79	1.80
Condenser Fan (kW)	0.67	0.66	0.67	0.67	0.66	0.67	0.66	0.66	0.66	0.66	0.66	0.67	0.67	0.66	0.65	0.66	0.66	0.66	0.66	0.66
Unit Performance																				
Total Cooling Capacity (Tons)	7.67	7.82	7.55	7.31	7.37	7.43	6.57	7.00	6.66	6.64	7.50	7.09	6.82	6.65	6.75	6.80	6.01	6.36	6.09	6.10
Sensible Cooling Factor	72%	72%	73%	75%	73%	74%	80%	77%	79%	79%	72%	75%	77%	78%	79%	78%	83%	82%	82%	82%
Coil Bypass Factor	29%	29%	28%	27%	27%	27%	24%	25%	25%	26%	30%	29%	27%	28%	27%	28%	26%	26%	26%	26%
Energy Efficiency Ratio (Btu/Wh)	8.4	8.7	8.3	7.9	7.9	8.0	6.9	7.4	7.0	7.0	8.1	7.5	7.1	6.9	7.0	7.1	6.1	6.5	6.2	6.2
Precooler Performance																				
Outlet Dry Bulb Temperature (°F)	89.1	86.5	87.9	91.8	91.8	89.8	97.6	95.3	96.7	96.7	92.9	95.1	98.7	100.4	99.8	98.2	104.6	103.0	103.9	103.9
Temperature Drop (°F)	16.0	18.5	17.1	13.3	13.2	15.3	7.3	9.6	8.3	8.3	22.2	19.8	16.4	14.6	15.4	16.8	10.4	12.1	11.2	11.4
Effectiveness	49%	50%	54%	52%	52%	55%	49%	51%	56%	55%	50%	54%	53%	50%	51%	56%	52%	51%	56%	56%
Water Use Rate (GPH)	12.5	13.2	11.9	10.7	10.4	11.5	8.0	8.5	7.8	7.9	15.3	13.2	12.2	11.7	12.5	12.0	9.1	10.1	9.3	9.1

"Max" Avg. Capacity:	7.30
"Max" Avg. Power:	11.3
"Max" Avg. EER:	7.78
"Max" Avg. Evap. Flow:	3,350
"Max" Avg. Cond. Flow:	4,720

Test Unit #2 - Baseline Unit with Evaporative Pre-cooler

Test Summary Information	Variable Condenser Resistance (IW)				Cycling Tests				
Nominal Test Conditions	0.00	-0.14	-0.25	-0.50	"C"				"D"
Condenser Inlet Dry Bulb (°F)	95	95	95	95	82	82	82	82	82
Condenser Inlet Wet Bulb (°F)	75	75	75	75	60	60	60	60	60
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	57	57	57	57	57
Evaporator External Resistance (IW)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
General									
Date	7/20/99	7/20/99	7/20/99	7/20/99	7/21/99	7/21/99	7/21/99	7/21/99	7/21/99
Start Time	3:31p	5:37p	4:11p	4:55p	12:52p	2:30p	3:37p	4:16p	4:54p
Duration (minutes)	30	30	34	30	30	61	35	31	42
Barometric Pressure (in. of Hg)	29.59	29.57	29.59	29.58	29.63	29.61	29.60	29.60	29.59
Condenser Air Properties									
Inlet Dry Bulb Temperature (°F)	95.0	95.0	95.0	95.0	82.1	82.4	82.5	82.6	83.1
Inlet Wet Bulb Temperature (°F)	75.1	75.0	75.6	75.8	62.5	63.2	63.1	63.2	65.0
Inlet Relative Humidity (%)	41%	41%	42%	43%	34%	35%	35%	35%	39%
Outlet Dry Bulb Temperature (°F)	110.3	108.0	106.9	104.3	96.4	96.0	95.2	94.3	95.2
Air Flow Rate (CFM)	4,710	5,230	5,540	6,300	4,790	4,740	4,760	4,780	4,770
Evaporator Air Properties									
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.1	79.8	79.4	80.5
Inlet Dew Point Temperature (°F)	60.2	60.1	60.2	60.1	47.1	49.5	48.8	49.3	49.4
Inlet Wet Bulb Temperature (°F) - calc.	66.9	66.8	66.9	66.8	60.5	61.6	61.2	61.3	61.7
Outlet Dry Bulb Temperature (°F)	61.4	61.5	61.6	61.6	55.9	57.9	57.9	58.7	59.9
Outlet Dew Point Temperature (°F)	56.3	56.2	56.3	56.2	46.8				
Air Flow Rate (CFM)	3,340	3,360	3,350	3,350	3,460	3,540	3,560	3,560	3,570
Power Consumption						↓ kWh / hr ↓			
Total Demand (kW)	10.85	10.60	10.52	10.30	9.88	7.63	5.05	2.64	2.11
Compressor (kW)	8.25	8.12	8.06	7.88	7.27				
Evaporator Blower (kW)	1.79	1.80	1.79	1.79	1.89				
Condenser Fan (kW)	0.67	0.65	0.64	0.60	0.66				
Unit Performance						↓ Ton-hr / hr (Sensible) ↓			
Total Cooling Capacity (Tons)	7.65	7.68	7.62	7.59	7.64	5.84	3.81	1.86	1.46
Sensible Cooling Factor	73%	74%	73%	73%					
Coil Bypass Factor	26%	27%	27%	28%					
Energy Efficiency Ratio (Btu/Wh)	8.6	8.7	8.7	8.8	9.3	9.2	9.1	8.5	8.3
Precooler Performance									
Outlet Dry Bulb Temperature (°F)	84.5	84.6	85.0	85.3	72.0	72.5	72.7	73.1	74.0
Temperature Drop (°F)	10.5	10.4	10.0	9.7	10.1	9.9	9.8	9.5	9.1
Effectiveness	53%	52%	52%	50%	52%	52%	50%	49%	50%
Water Use Rate (GPH)	8.9	9.3	9.6	10.0	8.8	7.4	5.5	5.4	5.7
		Cycle Period:			30	12	15	20	30
		"On" Fraction:			100%	75%	50%	25%	20%
		"On" Time (minutes):			30	9	7.5	5	6
		Cooling Load Factor:			100%	76.4%	49.9%	24.4%	19.1%
		Degradation Coefficient:				0.04	0.04	0.12	0.13
		SEER:							9.3

Test Unit #3 - High Efficiency Dual Compressor Unit

Test Summary Information	Low Temperature Tests			Low Evaporator Inlet Humidity ("Dry" Coil) Sensitivity Tests																	
Nominal Test Conditions			"Min2"																		
Condenser Inlet Dry Bulb (°F)	55	55	67	55	55	55	67	67	67	82	82	82	95	95	95	105	105	105	115	115	115
Evaporator Inlet Dry Bulb (°F)	55	67	67	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	46	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57	57
Evaporator External Resistance (IW)	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
General																					
Date	8/17/99	8/16/99	8/16/99	8/13/99	8/23/99	8/31/99	8/13/99	8/23/99	8/31/99	8/13/99	8/23/99	8/31/99	8/13/99	8/23/99	8/31/99	8/13/99	8/23/99	8/31/99	8/13/99	8/23/99	8/31/99
Start Time	5:38p	5:20p	3:58p	10:44a	8:52a	4:55p	12:14p	10:34a	5:53p	1:39p	12:20p	10:05a	2:57p	1:57p	1:16p	4:27p	3:29p	2:15p	5:54p	5:04p	3:24p
Duration (minutes)	30	30	30	31	30	30	30	30	30	30	30	30	31	30	30	30	30	30	30	30	31
Barometric Pressure (in. of Hg)	29.60	29.62	29.63	29.66	29.48	29.52	29.64	29.48	29.51	29.62	29.46	29.61	29.59	29.46	29.57	29.56	29.45	29.56	29.54	29.44	29.54
Condenser Air Properties																					
Inlet Dry Bulb Temperature (°F)	55.0	55.0	67.0	55.0	55.0	55.0	67.0	67.0	67.0	82.1	82.1	82.0	95.1	95.0	95.0	105.0	104.9	105.0	115.0	115.0	115.0
Inlet Relative Humidity (%)	62%	70%	56%	63%	68%	58%	61%	55%	49%	38%	36%	23%	26%	24%	18%	19%	16%	10%	12%	10%	5%
Outlet Dry Bulb Temperature (°F)	66.2	67.6	80.7	67.9	67.5	67.6	81.2	80.9	80.7	97.5	97.3	96.5	110.9	110.8	110.4	121.0	120.9	120.7	131.1	131.2	130.8
Air Flow Rate (CFM)	6,820	6,780	6,750	6,780	6,810	6,790	6,780	6,800	6,760	6,750	6,760	6,790	6,710	6,740	6,750	6,670	6,710	6,740	6,620	6,660	6,720
Evaporator Air Properties																					
Inlet Dry Bulb Temperature (°F)	55.0	67.0	67.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	35.8	50.0	49.5	46.2	46.0	40.4	48.1	47.7	45.0	49.5	48.7	38.6	50.9	49.7	42.7	51.9	50.1	40.2	52.8	51.2	41.5
Inlet Wet Bulb Temperature (°F) - calc.	45.7	57.0	56.7	60.2	60.1	57.9	61.0	60.7	59.6	61.6	61.2	57.4	62.2	61.6	58.8	62.6	61.8	57.9	63.1	62.3	58.3
Outlet Dry Bulb Temperature (°F)	38.1	50.8	50.0	58.3	58.0	57.4	57.4	56.9	56.6	56.9	56.3	56.2	57.4	56.9	57.0	58.3	57.8	58.1	59.6	59.3	59.4
Outlet Dew Point Temperature (°F)	32.9	45.4	45.1	44.8	44.8	40.1	47.0	46.7	44.6	48.8	48.1	38.5	50.3	49.4	42.6	51.4	50.0	40.1	52.6	51.0	41.4
Air Flow Rate (CFM)	2,840	2,980	2,970	3,000	2,960	3,030	2,980	2,950	3,000	2,980	2,910	3,020	2,980	2,950	3,030	2,980	2,960	3,030	2,980	2,990	3,030
Power Consumption																					
Total Demand (kW)	7.16	7.29	7.98	7.27	7.19	7.20	8.00	7.90	7.86	9.02	8.91	8.87	10.01	9.93	9.91	10.87	10.81	10.75	11.88	11.81	11.74
Compressor 1 (kW)	2.06	2.09	2.45	2.09	2.07	2.06	2.46	2.43	2.41	2.99	2.95	2.96	3.50	3.47	3.44	3.96	3.92	3.89	4.48	4.43	4.40
Compressor 2 (kW)	2.06	2.12	2.47	2.12	2.16	2.14	2.48	2.51	2.48	2.99	3.02	2.93	3.49	3.51	3.49	3.91	3.94	3.90	4.42	4.43	4.39
Evaporator Blower (kW)	2.40	2.44	2.43	2.41	2.32	2.36	2.41	2.33	2.34	2.41	2.32	2.36	2.40	2.34	2.37	2.39	2.35	2.36	2.38	2.35	2.36
Condenser Fan (kW)	0.64	0.65	0.63	0.64	0.64	0.63	0.64	0.63	0.63	0.63	0.62	0.62	0.62	0.61	0.61	0.61	0.60	0.60	0.59	0.59	0.59
Performance																					
Total Cooling Capacity (Tons)	5.14	6.01	6.13	6.39	6.32	6.31	6.52	6.52	6.53	6.56	6.51	6.61	6.37	6.30	6.39	6.06	6.01	6.07	5.63	5.65	5.68
Sensible Cooling Factor	88%	74%	76%	93%	94%	98%	95%	95%	98%	96%	97%	99%	97%	98%	100%	97%	99%	100%	98%	99%	100%
Coil Bypass Factor	27%	33%	29%	40%	39%	44%	32%	32%	34%	27%	26%	43%	24%	25%	39%	24%	26%	45%	26%	29%	47%
Energy Efficiency Ratio (Btu/Wh)	8.62	9.89	9.21	10.55	10.55	10.53	9.78	9.91	9.97	8.73	8.77	8.95	7.64	7.61	7.74	6.68	6.67	6.78	5.68	5.74	5.81

Test Unit #3 - High Efficiency Dual Compressor Unit

Test Summary Information	Moderate Evaporator Inlet Humidity SensitivityTests										
Nominal Test Conditions								-------------- "B2" ---------------			
Condenser Inlet Dry Bulb (°F)	55	55	55	55	67	67	67	82	82	82	82
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67
Evaporator External Resistance (IW)	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
General											
Date	8/12/99	8/20/99	8/24/99	9/2/99	8/12/99	8/20/99	9/2/99	8/20/99	8/25/99	9/1/99	9/2/99
Start Time	12:49p	8:26a	8:57a	1:54p	2:18p	10:01a	3:43p	11:12a	5:47p	12:25p	12:29p
Duration (minutes)	30	30	30	31	32	30	31	30	30	30	30
Barometric Pressure (in. of Hg)	29.69	29.62	29.49	29.47	29.66	29.62	29.45	29.62	29.38	29.44	29.48
Condenser Air Properties											
Inlet Dry Bulb Temperature (°F)	55.1	55.1	55.0	55.0	67.1	67.0	67.0	82.2	82.0	82.0	82.0
Inlet Relative Humidity (%)	81%	64%	72%	69%	66%	62%	52%	39%	29%	28%	34%
Outlet Dry Bulb Temperature (°F)	68.3	67.6	68.7	67.9	81.2	81.1	81.2	97.7	97.7	97.7	97.6
Air Flow Rate (CFM)	6,810	6,840	6,630	6,780	6,820	6,830	6,790	6,810	6,760	6,760	6,740
Evaporator Air Properties											
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.0	60.8	60.1	60.4	59.9	60.5	60.5	60.5	60.4	60.6	60.4
Inlet Wet Bulb Temperature (°F) - calc.	66.8	67.2	66.8	67.0	66.7	67.1	67.0	67.1	67.0	67.1	67.0
Outlet Dry Bulb Temperature (°F)	63.2	63.1	62.8	62.7	62.3	62.2	61.8	61.2	60.8	60.8	60.7
Outlet Dew Point Temperature (°F)	55.8	56.2	55.7	55.7	55.7	55.9	55.6	56.0	55.6	55.7	55.6
Air Flow Rate (CFM)	2,980	2,920	2,920	2,910	2,980	2,900	2,890	2,880	2,860	2,840	2,830
Power Consumption											
Total Demand (kW)	7.26	7.14	7.19	7.07	7.99	7.87	7.80	8.94	8.91	8.81	8.80
Compressor 1 (kW)	2.10	2.09	2.10	2.07	2.48	2.46	2.45	3.01	3.02	2.99	2.99
Compressor 2 (kW)	2.13	2.12	2.17	2.10	2.50	2.49	2.46	3.01	3.00	2.96	2.97
Evaporator Blower (kW)	2.38	2.28	2.29	2.26	2.38	2.28	2.26	2.29	2.27	2.24	2.23
Condenser Fan (kW)	0.65	0.64	0.64	0.64	0.64	0.64	0.63	0.62	0.61	0.62	0.61
Performance											
Total Cooling Capacity (Tons)	6.48	6.62	6.52	6.66	6.74	6.78	6.91	6.95	7.08	7.09	7.06
Sensible Cooling Factor	70%	68%	70%	68%	71%	69%	69%	71%	70%	69%	70%
Coil Bypass Factor	38%	37%	37%	37%	34%	33%	33%	28%	27%	27%	27%
Energy Efficiency Ratio (Btu/Wh)	10.72	11.12	10.88	11.31	10.12	10.33	10.64	9.33	9.54	9.66	9.63

"B2" Avg. Capacity:	7.05
"B2" Avg. Power:	8.9
"B2" Avg. EER:	9.54
"B2" Avg. Flow:	2,850

Test Unit #3 - High Efficiency Dual Compressor Unit

Test Summary Information	Moderate Evaporator Inlet Humidity SensitivityTests (Continued)														
Nominal Test Conditions	"A2"												"Max2"		
Condenser Inlet Dry Bulb (°F)	95	95	95	95	95	95	95	95	95	105	105	105	115	115	115
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67
Evaporator External Resistance (IW)	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
General															
Date	8/19/99	8/20/99	8/20/99	8/20/99	8/25/99	8/26/99	8/30/99	9/1/99	9/1/99	8/12/99	8/20/99	9/1/99	8/12/99	8/20/99	9/1/99
Start Time	1:58p	12:52p	4:22p	5:05p	11:45a	11:37a	1:52p	1:25p	4:42p	5:30p	2:08p	3:44p	6:38p	3:26p	2:40p
Duration (minutes)	30	30	21	27	41	30	32	30	32	30	30	31	32	26	30
Barometric Pressure (in. of Hg)	29.54	29.60	29.57	29.57	29.47	29.42	29.62	29.45	29.43	29.62	29.59	29.43	29.62	29.57	29.44
Condenser Air Properties															
Inlet Dry Bulb Temperature (°F)	95.1	95.0	94.9	95.0	95.0	95.0	95.0	95.0	95.0	105.0	105.0	105.0	115.0	115.0	115.0
Inlet Relative Humidity (%)	28%	27%	24%	24%	27%	24%	21%	19%	18%	19%	19%	9%	12%	12%	6%
Outlet Dry Bulb Temperature (°F)	111.5	111.7	111.5	111.5	111.6	111.4	111.2	111.5	111.8	121.8	121.9	121.9	132.0	132.1	132.0
Air Flow Rate (CFM)	6,730	6,720	6,670	6,680	6,710	6,730	6,750	6,700	6,660	6,700	6,700	6,650	6,660	6,670	6,660
Evaporator Air Properties															
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.6	60.5	60.5	60.5	60.7	60.4	60.7	60.3	60.3	59.9	60.4	60.4	60.0	60.7	60.5
Inlet Wet Bulb Temperature (°F) - calc.	67.1	67.1	67.0	67.0	67.1	67.0	67.1	66.9	66.9	66.7	67.0	67.0	66.8	67.2	67.0
Outlet Dry Bulb Temperature (°F)	61.1	61.1	61.0	61.1	60.9	60.8	61.0	60.6	60.7	61.4	61.3	61.0	62.0	62.1	61.7
Outlet Dew Point Temperature (°F)	56.3	56.3	56.2	56.1	56.2	56.0	56.2	55.9	55.9	56.4	56.6	56.3	57.1	57.4	57.0
Air Flow Rate (CFM)	2,880	2,870	2,880	2,880	2,830	2,850	2,860	2,850	2,850	2,960	2,890	2,860	2,960	2,890	2,860
Power Consumption															
Total Demand (kW)	9.88	9.91	9.90	9.91	9.86	9.85	9.83	9.80	9.82	10.95	10.84	10.67	11.96	11.86	11.64
Compressor 1 (kW)	3.51	3.52	3.53	3.53	3.53	3.50	3.49	3.50	3.52	4.02	3.99	3.95	4.54	4.51	4.45
Compressor 2 (kW)	3.47	3.51	3.49	3.49	3.48	3.49	3.47	3.46	3.46	3.96	3.96	3.88	4.45	4.46	4.36
Evaporator Blower (kW)	2.29	2.27	2.27	2.28	2.25	2.25	2.26	2.24	2.24	2.37	2.28	2.24	2.37	2.28	2.24
Condenser Fan (kW)	0.61	0.62	0.60	0.60	0.60	0.60	0.61	0.61	0.60	0.61	0.60	0.59	0.61	0.59	0.60
Performance															
Total Cooling Capacity (Tons)	6.85	6.86	6.89	6.90	6.88	6.90	6.95	6.97	6.95	6.55	6.61	6.74	6.17	6.24	6.32
Sensible Cooling Factor	72%	72%	72%	72%	71%	72%	71%	72%	72%	76%	74%	73%	78%	75%	75%
Coil Bypass Factor	25%	25%	25%	26%	25%	25%	26%	25%	25%	25%	24%	25%	25%	25%	24%
Energy Efficiency Ratio (Btu/Wh)	8.33	8.30	8.35	8.36	8.37	8.40	8.49	8.54	8.50	7.17	7.31	7.59	6.19	6.31	6.52

"A2" Avg. Capacity:	6.91	"Max2" Avg. Capacity:	6.24
"A2" Avg. Power:	9.9	"Max2" Avg. Power:	11.8
"A2" Avg. EER:	8.40	"Max2" Avg. EER:	6.34
"A2" Avg. Flow:	2,860	"Max2" Avg. Flow:	2,910

Test Unit #3 - High Efficiency Dual Compressor Unit

Test Summary Information	High Evaporator Inlet Humidity Sensitivity Tests														
Nominal Test Conditions															
Condenser Inlet Dry Bulb (°F)	55	55	67	67	82	82	95	95	95	105	105	105	115	115	115
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Evaporator External Resistance (IW)	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
General															
Date	8/17/99	8/27/99	8/24/99	8/27/99	8/17/99	8/27/99	8/17/99	8/25/99	8/27/99	8/19/99	8/25/99	8/27/99	8/19/99	8/25/99	8/27/99
Start Time	10:36a	12:13a	5:37p	8:13a	1:39p	7:16a	3:08p	7:46a	6:05a	9:47a	6:44a	4:54a	11:08a	5:31a	3:31a
Duration (minutes)	36	30	30	35	33	30	33	30	30	30	30	30	31	30	30
Barometric Pressure (in. of Hg)	29.67	29.44	29.43	29.44	29.64	29.44	29.62	29.49	29.42	29.58	29.48	29.42	29.57	29.46	29.42
Condenser Air Properties															
Inlet Dry Bulb Temperature (°F)	55.0	55.0	67.0	67.0	82.0	82.0	94.9	95.1	95.0	105.3	105.0	105.0	115.1	115.0	115.0
Inlet Relative Humidity (%)	68%	69%	60%	57%	40%	45%	22%	26%	29%	20%	18%	20%	13%	13%	14%
Outlet Dry Bulb Temperature (°F)	68.1	68.0	82.2	81.6	98.0	98.1	112.1	113.3	112.2	123.1	124.0	122.9	133.7	134.4	133.3
Air Flow Rate (CFM)	6,820	6,810	6,630	6,760	6,780	6,730	6,740	6,440	6,700	6,740	6,400	6,690	6,690	6,430	6,720
Evaporator Air Properties															
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.8	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	73.1	73.3	72.8	73.2	72.9	73.1	73.0	73.5	73.2	73.0	72.8	73.2	73.1	72.4	73.2
Inlet Wet Bulb Temperature (°F) - calc.	75.0	75.1	75.0	75.0	74.8	75.0	74.9	75.3	75.0	74.9	74.8	75.0	75.0	74.5	75.0
Outlet Dry Bulb Temperature (°F)	69.6	69.3	69.1	68.8	69.0	68.7	69.0	69.2	68.8	69.2	68.9	69.0	69.6	69.1	69.4
Outlet Dew Point Temperature (°F)	66.9	66.8	66.0	66.1	65.9	65.8	65.8	66.2	65.7	65.8	65.7	65.6	66.1	65.6	66.0
Air Flow Rate (CFM)	2,910	2,830	2,850	2,820	2,900	2,810	2,880	2,830	2,790	2,870	2,840	2,800	2,870	2,860	2,820
Power Consumption															
Total Demand (kW)	7.19	7.04	7.93	7.77	9.01	8.83	10.07	10.01	9.87	10.96	10.93	10.75	11.96	12.00	11.80
Compressor 1 (kW)	2.10	2.08	2.51	2.48	3.06	3.03	3.61	3.60	3.57	4.07	4.08	4.03	4.59	4.60	4.54
Compressor 2 (kW)	2.13	2.12	2.56	2.48	3.03	2.99	3.56	3.59	3.51	4.03	4.04	3.94	4.52	4.58	4.47
Evaporator Blower (kW)	2.31	2.20	2.23	2.20	2.30	2.19	2.29	2.21	2.18	2.25	2.22	2.18	2.25	2.23	2.19
Condenser Fan (kW)	0.65	0.63	0.62	0.62	0.63	0.61	0.61	0.60	0.60	0.61	0.59	0.59	0.60	0.59	0.60
Performance															
Total Cooling Capacity (Tons)	6.80	6.84	7.25	7.21	7.27	7.36	7.38	7.31	7.41	7.31	7.25	7.37	7.09	6.98	7.09
Sensible Cooling Factor	40%	39%	41%	39%	39%	38%	38%	37%	38%	38%	39%	37%	38%	40%	37%
Coil Bypass Factor	55%		52%												
Energy Efficiency Ratio (Btu/Wh)	11.4	11.7	11.0	11.1	9.7	10.0	8.8	8.8	9.0	8.0	8.0	8.2	7.1	7.0	7.2

Test Unit #3 - High Efficiency Dual Compressor Unit

Test Summary Information	Integrated Part Load Value Tests							Variable Evaporator Resistance Tests					Variable Condenser Resistance Tests				Cycling Performance Tests					
Nominal Test Conditions	"IPLV2"		------ "IPLV1" ------			"B1"	"A1"					(IW) →	0.00	0.10	0.20	-0.03	"C21"	"D21"	"C1"	"D1"	"C2"	"D2"
Condenser Inlet Dry Bulb (°F)	80	80	80	80	80	82	95	95	95	95	95	95	95	95	95	95	82	82	82	82	82	82
Evaporator Inlet Dry Bulb (°F)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Evaporator Inlet Wet Bulb (°F)	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	67	57	57	57	57	57	57
Evaporator External Resistance (IW)	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.40	0.60	0.70	0.80	1.00	0.70	0.70	0.70	0.70	0.70	0.72	0.70	0.72	0.70	0.70
General																						
Date	8/16/99	8/26/99	8/19/99	8/26/99	8/26/99	8/26/99	8/26/99	8/19/99	8/19/99	8/19/99	8/19/99	8/19/99	8/26/99	8/26/99	8/26/99	8/26/99	8/31/99	8/31/99	9/1/99	9/1/99	9/2/99	9/2/99
Start Time	12:59p	10:26a	8:17a	5:11a	6:38a	5:52a	7:51a	4:21p	2:36p	1:58p	3:14p	3:44p	11:37a	12:26p	1:19p	3:30p	10:05a	11:00a	8:55a	9:50a	9:15a	10:10a
Duration (minutes)	31	31	31	30	30	30	50	18	30	30	20	20	30	30	30	30	30	60	30	120	30	90
Barometric Pressure (in. of Hg)	29.64	29.43	29.58	29.40	29.41	29.40	29.44	29.52	29.54	29.54	29.53	29.52	29.42	29.41	29.41	29.38	29.61	29.60	29.47	29.47	29.48	29.49
Condenser Air Properties																						
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	82.0	95.2	95.0	95.1	95.1	95.0	95.0	95.0	95.1	95.0	95.0	82.0	82.5	82.0	82.4	82.0	82.0
Inlet Relative Humidity (%)	42%	41%	42%	39%	38%	37%	24%	22%	28%	28%	28%	26%	24%	24%	24%	28%	23%	21%	32%	29%	36%	34%
Outlet Dry Bulb Temperature (°F)	95.7	95.6	87.4	87.7	87.6	89.7	103.2	111.6	111.5	111.5	111.3	111.0	111.4	116.0	123.3	110.9	96.5	95.8	89.3	88.9	97.3	96.4
Air Flow Rate (CFM)	6,740	6,740	6,780	6,740	6,750	6,740	6,720	6,740	6,740	6,730	6,730	6,740	6,730	5,480	4,140	6,930	6,790	6,470	6,750	6,480	6,730	6,730
Evaporator Air Properties																						
Inlet Dry Bulb Temperature (°F)	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	79.9	80.0	80.0	80.0	80.0
Inlet Dew Point Temperature (°F)	60.1	60.5	60.2	60.8	60.3	60.2	60.6	59.9	60.8	60.6	60.4	59.6	60.4	60.5	60.2	60.7	38.6	37.1	48.1	45.9	51.6	50.0
Inlet Wet Bulb Temperature (°F) - calc.	66.8	67.0	66.9	67.2	66.9	66.9	67.1	66.7	67.2	67.1	67.0	66.5	67.0	67.1	66.8	67.1	57.4	56.8	60.9	60.0	62.5	61.8
Outlet Dry Bulb Temperature (°F)	61.5	61.1	71.6	71.7	71.6	71.6	72.0	61.9	61.7	61.1	60.5	59.2	60.8	60.5	60.5	61.0	56.2	59.6	69.0	70.2	56.8	58.8
Outlet Dew Point Temperature (°F)	55.8	55.9	57.9	58.1	57.8	57.7	58.2	56.6	56.9	56.3	55.8	54.6	56.0	56.1	56.1	56.3	38.5	37.0	47.7	45.7	49.9	48.6
Air Flow Rate (CFM)	2,950	2,860	2,930	2,900	2,900	2,900	2,900	3,280	3,020	2,880	2,750	2,480	2,850	2,850	2,870	2,850	3,020	3,030	2,970	2,990	2,880	2,950
Power Consumption																			↓ kWh / hr ↓			
Total Demand (kW)	8.90	8.72	5.84	5.78	5.79	5.86	6.36	10.08	9.96	9.88	9.83	9.70	9.85	10.20	10.85	9.84	8.87	1.84	5.89	1.25	8.79	6.46
Compressor 1 (kW)	2.96	2.92	2.92	2.93	2.93	3.01	3.51	3.52	3.52	3.51	3.51	3.49	3.50	3.68	3.99	3.49	2.96		2.99		2.96	
Compressor 2 (kW)	2.95	2.92	0.00	0.00	0.00	0.00	0.00	3.49	3.48	3.47	3.47	3.46	3.49	3.67	3.99	3.48	2.93		0.00		2.95	
Evaporator Blower (kW)	2.37	2.26	2.27	2.21	2.22	2.22	2.22	2.46	2.35	2.29	2.24	2.14	2.25	2.26	2.26	2.26	2.36		2.27		2.27	
Condenser Fan (kW)	0.63	0.61	0.64	0.63	0.63	0.63	0.62	0.61	0.61	0.61	0.61	0.61	0.60	0.60	0.62	0.60	0.62		0.64		0.61	
Performance																			↓ Ton-hr / hr (Sensible) ↓			
Total Cooling Capacity (Tons)	6.90	6.96	3.28	3.35	3.32	3.31	3.16	7.03	6.90	6.85	6.78	6.58	6.90	7.01	6.86	6.84	6.58	1.21	2.91	0.56	6.11	3.47
Sensible Cooling Factor	72%	70%	67%	64%	65%	66%	65%	76%	72%	72%	71%	71%	72%	72%	74%	71%						
Coil Bypass Factor	29%	28%	71%	72%	72%	72%	73%	26%	26%	25%	25%	23%	25%	23%	23%	25%	43%		67%		24%	
Energy Efficiency Ratio (Btu/Wh)	9.3	9.6	6.7	7.0	6.9	6.8	6.0	8.4	8.3	8.3	8.3	8.1	8.4	8.2	7.6	8.3	8.9	7.9	5.9	5.4	8.3	6.4
Percent of full load capacity (avg.):					47.9%																	
Part Load Factor (PLF):					0.385																	
Integrated Part Load Value (IPLV):					7.66																	
Cycle Period:																	30	30	30	30	30	30
"On" Fraction:																	100%	20%	100%	20%	100%	20%
"On" Time (minutes):																	30	6	30	6	30	6
Cooling Load Factor:																	100%	18.4%	100%	19%	100%	56.8%
Degradation Coefficient:																		0.139		0.111		0.526
SEER:																		8.9		9.0		7.0
																	Cycling both Compressors		Only Cycing Compressor 1		Only Cycing Compressor 2	

APPENDIX V

Test Unit Specifications

Test Unit #1: Standard Efficiency Baseline Unit

ARI Net Cooling Capacity	86,000 Btu/hr	7.2 tons
System Power	9.66 kW	
EER	8.9	
Supply Air Flow (cfm)	3000 Nominal	2625 ARI Rated
Evaporator Inlet Temperatures (rated conditions)	80°F dry bulb	67°F
Condenser (Outdoor) Temperatures (rated conditions)	95°F dry bulb	
* Note: this is a high heat model		
Number of Compressors	1	
Compressor Type	Reciprocating	
Indoor Coil Face Area (ft²)	7.88	
Refrigerant Control	Orifice	
Indoor Fan Type	Centrifugal	
Indoor Fan Motor HP	1.0 standard	2.0 high heat model
Outdoor Coil Face Area (ft²)	12.09	
Outdoor Fan Type (number)	Propeller (1)	
Outdoor Fan Motor HP	0.5	
Outdoor Fan Nominal Air Flow (cfm)	5450	

Test Unit #2: Standard Efficiency Unit with Evaporative Condenser Pre-cooler

Same physical specifications as listed above for Test Unit #1, with the following additions:

Evaporative Pad Thickness	3”	
Exposed Evaporative Pad Area	approximately 11.0 (ft^2)	1.5 ft^2/ton capacity
Effectiveness (for face velocities between about 400 to 500 ft/min)	60%	
Pressure drop (in wc)	0.04” at 450 ft/min	
Water pump	1/60 hp	230v, 0.5a
Estimated Performance Factors:		
Evaporator Inlet Temperatures (rated conditions)	80°F dry bulb	67°F wet bulb
Condenser (Outdoor) Temperatures (rated conditions)	95°F dry bulb	75°F wet bulb
Calculated Condenser Inlet Temperature (for 60% effectiveness)	83°F dry bulb	
Estimated Cooling Capacity at 83°F dry bulb condenser inlet temperature	93,600 Btu/hr	7.8 tons
Estimated System Power	9.48 kW	
Estimated EER	9.9 Btu/Wh	

Test Unit #3: High Efficiency Dual Compressor Unit

ARI Net Cooling Capacity	90,000 Btu/hr	7.5 tons
System Power	8.18 kW	
EER	11.0	
IPLV	11.6	
Supply Air Flow (cfm)	3000 Nominal	
Evaporator Inlet Temperatures (rated conditions)	80°F dry bulb	67°F
Condenser (Outdoor) Temperatures (rated conditions)	95°F dry bulb	
* Note: this is a high heat model		
Number of Compressors	2	
Compressor Type	Scroll	
Indoor Coil Face Area (ft²)	8.9	
Refrigerant Control	Orifice	
Indoor Fan Type	Centrifugal	
Indoor Fan Motor HP	2.9 max continuous bhp	
Outdoor Coil Face Area (ft²)	20.5	
Outdoor Fan Type (number)	Propeller (2)	
Outdoor Fan Motor HP	0.25	
Outdoor Fan Nominal Air Flow (cfm)	6500	

APPENDIX VI

Uncertainty Analysis

Uncertainty Analysis for Laboratory Performance Testing of Packaged Air Conditioning Units

Introduction

The performance factors determined during these tests were primarily cooling capacity, power use, and energy efficiency ratio. The uncertainty in these performance factors due to measurement system errors is due primarily to uncertainties in the temperature, pressure, humidity, flow, and power measurements. These include bias errors due to instrument calibration and measurement locations (spatial errors), and precision errors due random fluctuations in the measurements (due to either instrument fluctuations or actual process fluctuations).

To perform an uncertainty analysis, the uncertainties in the individual measurements must be estimated. Then, the sensitivity of the final result to those measurements must be determined. Finally, the contributions of each measurement uncertainty must be combined to give an overall uncertainty in the result. Standard statistical techniques for uncertainty analysis were used, such as those described in ASME Power Test Code 19.1-1998 “Test Uncertainty” (ASME 1998), and Ronald Dieck’s “Measurement Uncertainty, Methods and Applications (Dieck, 1992).

Performance Analysis Method

The primary calculation of cooling capacity is done using the Indoor Air-Enthalpy Method described in Appendix II. This method calculates the cooling performed on the air passing over the evaporator coil. In summary, the calculation is:

$$q_{tci} = 60\ Q_{mi}\ (h_{a1} - h_{a2}) / [v'_n\ (1 + W_n)]$$

where,

q_{tci} = total cooling calculated with indoor-side data, Btu/hr
Q_{mi} = indoor-side actual volumetric flow rate of air-water mixture, acfm
h_{a1}, h_{a2} = air enthalpy entering and exiting the evaporator coil, respectively, Btu/lb dry air
v'_n = specific volume of air-water mixture at the indoor-side flow nozzle, ft³/lb
W_n = humidity ratio at the indoor-side flow nozzle, lb moisture per lb dry air

Various measurements go into determining each of these parameters. These are summarized below:

Parameter:	Function of:
Air mass flow (Q_{mi}, v'_n, W_n)	static pressure
	dry bulb temperature
	dew point temperature (or wet bulb/dry bulb temperatures)
	differential pressure across flow nozzles
	nozzle throat diameters
	nozzle calibration coefficient
Air enthalpy (h_{a1}, h_{a2})	static pressure
	dry bulb temperature
	dew point temperature (or wet bulb/dry bulb temperatures)

As a secondary check, the Outdoor Air Enthalpy Method was used. This method utilizes a heat balance on the unit, and calculates the cooling performed by subtracting the electrical energy input (in thermal units) from the heat rejected by the condenser. In summary, the calculation is:

$$q_{tco} = 60\, Q_{mo}\, (h_{a4} - h_{a3}) / [v'_n\, (1 + W_n)] - 3.41\, E_t$$

where,

q_{tco} = total cooling, calculated with outdoor-side data, Btu/hr
Q_{mo} = outdoor-side actual volumetric flow rate of air-water mixture, acfm
h_{a4}, h_{a3} = air enthalpy entering and exiting the condenser, respectively, Btu/lb dry air
v'_n = specific volume of air-water mixture at the outdoor-side flow nozzle, ft^3/lb
W_n = humidity ratio at the outdoor-side flow nozzle, lb moisture per lb dry air
E_t = total power to the unit, watts

The measurements which go into determining these parameters are analogous to those listed above for the indoor-side calculations, with the addition of the electric power measurement.

The energy efficiency ratio (EER) is the ratio of the cooling capacity to the total electrical demand:

$EER = q_{tc}/E_t$, Btu/Wh

Test Unit #2 had one additional performance factor calculated: the effectiveness of the evaporative cooling pad. This is calculated as the ratio of the measured dry bulb temperature drop across the evaporative cooler to the inlet wet bulb depression (difference between the dry bulb and wet bulb temperatures).

$$\varepsilon = \frac{t_{DB,in} - t_{DB,out}}{t_{DB,in} - t_{WB,in}} \qquad \text{evaporative cooler effectiveness}$$

During testing, online calculations of the above performance parameters were performed to check for stability and reasonableness of the data. Final data analysis was performed in a spreadsheet after the raw data had been averaged over the test periods. Air properties were determined using a computer program which uses standard ASHRAE relationships.

Uncertainty Analysis Method

The uncertainty in a result of a measurement or combination of measurements can be estimated by the following equation:

$$U_R = [\, B_R^2 + (t_{95} * S_R)^2\,]^{1/2}$$

where,

B_R =combined Bias uncertainty in the result
S_R = combined precision index in the result
t_{95} = Student's t for 95% confidence

The uncertainty in a calculated result (either the bias or precision component) can be determined using a Taylor series approximation as follows:

$$U_R^2 = [\, (U_{p1}\, \delta R/\delta p1)^2 + (U_{p2}\, \delta R/\delta p2)^2 + \ldots + (U_{pn}\, \delta R/\delta pn)^2\,]$$
$$+ [2*(\delta R/\delta p1)*(\delta R/\delta p2)*\rho_{12}* U_{p1}* U_{p2} + 2*(\delta R/\delta p2)*(\delta R/\delta p3)*\rho_{23}* U_{p2}* U_{p3} + \ldots\ldots.]$$

where,

U_R = absolute uncertainty in the result (either bias or precision)

U_{p1} = absolute uncertainty in parameter #1

$\delta R/\delta p1$ = partial derivative of the result with respect to parameter #1, etc., for all parameters up to parameter # n

ρ_{12} = correlation coefficient between parameters #1 and #2, etc., for all parameters up to n

The first set of bracketed terms are used by themselves if all error sources are independent. The second set of bracketed terms, sometimes called "cross-product" terms, are required if there are correlations between error sources.

The partial derivatives are also referred to as "sensitivity coefficients", and can be designated as:

$\Theta_{p1} = \delta R/\delta p1$

where,

Θ_{p1} = sensitivity coefficient of the result to parameter #1

For this analysis, it was assumed all error sources were independent. Therefore, the correlation coefficients are zero and the cross-product terms drop out of the equation.

Therefore, the final form of the uncertainty equation is:

$U_R^2 = [\, (U_{p1} \cdot \Theta_{p1})^2 + (U_{p2} \cdot \Theta_{p2})^2 + \ldots + (U_{pn} \cdot \Theta_{pn})^2 \,]$

The sensitivity coefficients, Θ_p, can be determined by taking partial derivatives of the equations for cooling capacity and EER, or they can be determined by "dithering". Dithering is a numerical technique used to estimate the sensitivity coefficients by changing each parameter a small amount and recalculating the result. The change in result divided by the change in the parameter is the sensitivity coefficient. This is especially useful for complex calculations, where partial differentiation is difficult. In this case, expressing enthalpies and other air properties explicitly in terms of the measured parameters and performing the partial differentiation would get complicated, so the dithering approach was used. Using actual test data, each measurement was varied a small amount, and the resulting change in the desired performance factor was determined. The change in the performance factor divided by the change in the varied parameter is the sensitivity coefficient.

Results

Because of the range of operating conditions applied during this testing, the sensitivity coefficients were not constant. In other words, the effect of a measurement on the overall uncertainty in the result could be more or less depending on the specific operating conditions. For this analysis, the sensitivity coefficients were determined for several test conditions on each unit, in order to bracket the range of uncertainties expected.

Table VI-1 below gives a summary of the uncertainties in the cooling capacity, power, and EER for Test Unit #1 for several of the test conditions.

Table VI-1: Summary of Selected Uncertainty Analysis Results for Test Unit #1

Test Designation	A	B	C	Max
Nominal Test Conditions:				
Indoor entering air dry bulb temperature (°F)	80	80	80	80
Indoor entering air wet bulb temperature (°F)	67	67	57	67
Outdoor entering air dry bulb temperature (°F)	95	82	82	115
Cooling Capacity Result (tons)	7.40	8.03	7.52	6.33
Cooling Capacity Uncertainty (±%)	4.0%	3.7%	3.6%	4.7%
Power Result (kW)	11.1	10.4	10.2	12.1
Power Uncertainty (±%)	1%	1%	1%	1%
EER Result (Btu/Wh)	8.02	9.28	8.85	6.26
EER Uncertainty (±%)	4.1%	3.8%	3.7%	4.8%

This table shows that the uncertainty in the cooling capacity and EER varied between about ±4% and ±5%, depending on the test conditions. The power measurement uncertainty was ±1%.

As an example, Table VI-2 below shows the complete uncertainty analysis for Test Unit #1 for the standard ARI rating test “A” conditions (80°F dry bulb/67°F wet bulb indoor air entering temperatures; 95°F dry bulb outdoor air entering temperature). This shows all of the details that went into determining the uncertainty values summarized above.

Table VI-2: Test Unit #1 Uncertainty Analysis Results for "A" Test Conditions

Nominal Test Condition (IDB/IWB - ODB) Test # Test Unit #1	see Note 4	80/67 - 95 0615E								
Parameter	Units	Test Value	Sensitivity Coefficient (tons per unit parameter change)	Parameter Bias Uncertainty	Cooling Capacity Bias Uncertainty (tons)	Cooling Capacity Bias Uncertainty (%)	Parameter Standard Deviation	Parameter Precision Index	Cooling Capacity Precision Index (tons)	Cooling Capacity Precision Index (%)
Indoor Side (Evaporator Coil)										
Barometric Pressure	psia	14.51	0.1159	0.05	0.006	0.08%				
Entering Air Dry Bulb Temp	° F	80.0	0.2959	0.30	0.089	1.21%	0.30	0.0224	0.0066	0.08%
Entering Air Dew Point Temp	° F	60.2	0.5427	0.30	0.163	2.22%	0.50	0.0373	0.0202	0.25%
Leaving Air Dry Bulb Temp	° F	61.8	-0.2950	0.50	-0.148	-2.01%	0.30	0.0224	-0.0066	-0.08%
Leaving Air Dew Point Temp	° F	56.3	-0.4778	0.30	-0.143	-1.96%	0.50	0.0373	-0.0178	-0.22%
Nozzle Flow Diff Pressure	in wc	2.294	1.5789	0.02	0.032	0.43%	0.04	0.0030	0.0047	0.06%
Nozzle Flow Static Pressure	in wc	-0.099	0.0092	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	62.3	-0.0071	0.50	-0.004	-0.05%				
Nozzle Diameters (2)	in	see note 1	2.0778	0.01	0.021	0.28%				
Nozzle Coefficient	%	see note 2	0.0733	1.00	0.073	1.00%				
Air Flow	acfm	3265								
Cooling Capacity	tons	7.33			0.29	3.94%			0.03	0.39%
EER	Btu/Wh	7.98			0.32	4.07%			0.03	0.39%
Overall Cooling Capacity Uncertainty	**%**	**4.0%**								
Overall EER Uncertainty	**%**	**4.1%**								
Outdoor Side (Condenser Coil)										
Barometric Pressure	psia	14.51	0.3323	0.05	0.017	0.23%				
Entering Air Dry Bulb Temp	° F	95.0	-0.4259	0.80	-0.341	-4.65%	0.30	0.0224	-0.0095	-0.13%
Entering Air Wet Bulb Temp	° F	67.8	0.0000	0.80	0.000	0.00%	0.50	0.0373	0.0000	0.00%
Leaving Air Dry Bulb Temp	° F	118.1	0.4262	1.50	0.639	8.72%	0.30	0.0224	0.0095	0.13%
Leaving Air Sample Dry Bulb Temp	° F	109.8	-0.0042	0.50	-0.002	-0.03%	0.40	0.0298	-0.0001	0.00%
Leaving Air Sample Wet Bulb Temp	° F	72.6	0.0097	0.50	0.005	0.07%	0.40	0.0298	0.0003	0.00%
Nozzle Flow Diff Pressure	in wc	0.893	5.3900	0.02	0.108	1.47%	0.01	0.0009	0.0050	0.07%
Nozzle Flow Static Pressure	in wc	-0.062	0.0124	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	117.5	-0.0087	0.50	-0.004	-0.06%				
Nozzle Diameters (3)	in	see note 1	2.2178	0.02	0.044	0.61%				
Nozzle Coefficient	%	see note 2	0.0985	2.00	0.197	2.69%				
Total Power	kW	11.03	-0.2844	0.1103	-0.031	-0.43%				
Air Flow	acfm	5207								
Cooling Capacity (Note 3)	tons	6.71			0.76	11.33%			0.01	0.21%
Overall Cooling Capacity Uncertainty	%	11.3%								
Heat Balance Error	%	8.4%								
Uncertainty in Heat Balance	%	12.0%								

Note 1: The indoor room air flow chamber used two nozzles with following diameters: 7.991, 6.002 inches
The outdoor room air flow chamber used three nozzles with the following throat diameters: 8.998, 8.997, 8.991 inches
Note 2: Nozzle coefficient uncertainty was used to account for possible flow profile effects.
Note 3: Cooling Capacity from outdoor side calculations uses sensible energy change through condenser

Note 4: IDB = indoor dry bulb
IWB = indoor wet bulb
ODB = outdoor dry bulb

This table first gives an example of measured test values for one of the condition "A" tests. Next it shows the sensitivity coefficients for each of these measured parameters, and the assumed bias uncertainties for each parameter. Using the sensitivity coefficients and the parameter bias uncertainties, the resulting bias uncertainties in the cooling capacity calculation are the shown (both in absolute tons and percent of the total cooling tons performed). The last four columns deal with the precision uncertainties, and are based on typical standard deviations of the measured parameters over time. Typically, 30 minutes of data were collected for each test point, and with data storage intervals of 10 seconds, this resulted in 180 values being average for each test point. This was used to calculate the precision index for each parameter.

The cooling capacity bias and precision uncertainties were combined separately, and then combined together to give an overall cooling capacity uncertainty. This is listed below the data for the indoor side parameters, and is in bold font. Similarly, the EER uncertainty combines the cooling capacity uncertainty and the power measurement uncertainty (1%) to give the overall EER uncertainty.

The bias uncertainties in the entering and leaving dry bulb temperature measurements include estimates of the spatial error. The spatial error is due to having a limited number of sensors (4 in this case) to measure an average temperature in the supply and return ducts. The uncertainty is higher for the leaving air temperature because there is more stratification in this duct.

The indoor side data are used for the primary performance calculations. These are the most accurate, with a cooling capacity uncertainty of 4.0% and an EER uncertainty of 4.1% in this example. The major contributions to the overall uncertainty were fairly evenly spread between the entering air dew point, the leaving air dry bulb, the leaving air dew point, and the flow measurement. The entering air dry bulb also contributed a significant amount. A 1% uncertainty in the overall flow nozzle coefficient was assumed as a worst case to account for possible flow profile effects and nozzle coefficient variations. The pressure measurement effects on the flow uncertainty were relatively small.

This table also shows the uncertainties calculated on the outdoor side. The outdoor side data were used for a heat balance check, so the uncertainties were estimated in order to determine a reasonable heat balance error. The uncertainty on the outdoor side was relatively large, with a value of 11.3% in this example. It should be noted that this uncertainty is due to measurement error only, and does not account for heat losses from the unit which would add to the heat balance error. The primary reason for the larger outdoor side cooling capacity uncertainty is the large uncertainty in the condenser outlet temperature measurement. This was due to two factors. First, there was a large temperature variation across the outlet duct, resulting in a large spatial uncertainty. Second, twelve thermocouples were used in this location to attempt to get a better average. The thermocouples are not as accurate as RTDs, and post-test calibration checks showed a drift of about 0.6°F. These effects combined to give an overall estimated uncertainty in this measurement of about 1.5°F. There is also a significant spatial error in the condenser inlet temperature measurement. Flow errors on the outdoor side also contribute a relatively higher uncertainty to the cooling capacity calculation than on the indoor side. A higher uncertainty in the flow coefficient (2%) was also assumed because of the ducting arrangement on the outdoor-side flow chamber. These effects combine to give the outdoor side cooling capacity uncertainty of greater than 11%.

Combining the uncertainties on the indoor and outdoor sides results in an uncertainty in the heat balance calculation of about 12% in this example. The actual heat balance error, calculated as the percent difference between the indoor-side and outdoor-side cooling capacity results, was 8.4% in this example. This is within the uncertainty in the heat balance error, which indicates that the assumed parameter uncertainties are reasonable. In general, the heat balance error varied from about 5% to 15% depending on the conditions, and was typically within the calculated uncertainty in the heat balance. The drift in the condenser outlet temperature measurement was in a direction (reading low) to account for about 4% (absolute) of the heat balance error.

Similar analysis was performed for Test Unit #2 and Test Unit #3. The uncertainty in the cooling capacity and EER for these units ranged from about ±3% to ±4%. The power uncertainty remained at ±1%. There was a slight improvement in the uncertainty on the indoor-side because of several changes made in between tests. The supply and return ducts to the test unit were better insulated around the temperature measurement locations to reduce the spatial uncertainty due to variations between the probes. Also, improvements were made to the dew point sampling system for the supply and return moisture conditions. Additional fans were placed around the test unit to improve the mixing of air entering the condenser. The uncertainty results are summarized below:

Table VI-3: Summary of Selected Uncertainty Analysis Results for Test Unit #2

Test Designation	A	B	C	Max
Nominal Test Conditions:				
Indoor entering air dry bulb temperature (°F)	80	80	80	80
Indoor entering air wet bulb temperature (°F)	67	67	57	67
Outdoor entering air dry bulb temperature (°F)	95	82	82	115
Outdoor entering air wet bulb temperature (°F)	75	65	65	75
Cooling Capacity Result (tons)	7.69	8.30	7.77	7.27
Cooling Capacity Uncertainty (±%)	3.7%	3.4%	3.1%	4.0%
Power Result (kW)	10.7	10.1	9.88	9.43
Power Uncertainty (±%)	1%	1%	1%	1%
EER Result (Btu/Wh)	8.58	9.90	9.43	8.30
EER Uncertainty (±%)	3.8%	3.5%	3.3%	4.1%

For Test Unit #2, an additional performance factor was calculated. This was the effectiveness of the evaporative cooling pads. Because of the difficulty in measuring the dry bulb temperature at the exit of the evaporative pads, the effectiveness calculation is relatively inaccurate. A grid of thermocouples was used to measure this temperature, but there was a relatively large variation in this temperature across the outlet of the evaporative cooler. Therefore, obtaining an accurate average was difficult. With an inlet dry bulb and wet bulb temperature uncertainty of about 0.6°F, and an outlet dry bulb temperature uncertainty of 1.5°F, the uncertainty in the effectiveness is about ±6% points. As given in the results section of this report, the effectiveness was found to average about 51%, with a range of 46% to 56%. The calculated uncertainty is in relative agreement with the observed range.

Below is an example of the uncertainty analysis details for Test Unit #2 for Test Condition “A”.

Table VI-4: Test Unit #2 Uncertainty Analysis for "A" Test Conditions

Nominal Test Condition (IDB/IWB - ODB/OWB) Test # Test Unit #2	see Note 4	80/67 - 95/75 0720E								
Parameter	Units	Test Value	Sensitivity Coefficient	Parameter Bias Uncertainty	Cooling Capacity Bias Uncertainty		Parameter Standard Deviation	Parameter Precision Index	Cooling Capacity Precision Index	
			(tons per unit parameter change)		(tons)	(%)			(tons)	(%)
Indoor Side (Evaporator Coil)										
Barometric Pressure	psia	14.53	0.1226	0.05	0.006	0.08%				
Entering Air Dry Bulb Temp	° F	80.0	0.3034	0.30	0.091	1.19%	0.30	0.0224	0.0068	0.09%
Entering Air Dew Point Temp	° F	60.2	0.5554	0.30	0.167	2.18%	0.50	0.0373	0.0207	0.27%
Leaving Air Dry Bulb Temp	° F	61.4	-0.3025	0.40	-0.121	-1.58%	0.30	0.0224	-0.0068	-0.09%
Leaving Air Dew Point Temp	° F	56.3	-0.4884	0.30	-0.147	-1.91%	0.50	0.0373	-0.0182	-0.24%
Nozzle Flow Diff Pressure	in wc	2.406	1.5725	0.02	0.031	0.41%	0.04	0.0030	0.0047	0.06%
Nozzle Flow Static Pressure	in wc	-0.118	0.0096	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	62.0	-0.0075	0.50	-0.004	-0.05%				
Nozzle Diameters (2)	in	see note 1	2.1693	0.01	0.022	0.28%				
Nozzle Coefficient	%	see note 2	0.0765	1.00	0.077	1.00%				
Air Flow	acfm	3338								
Cooling Capacity	tons	7.65			0.28	3.69%			0.03	0.39%
EER	Btu/Wh	8.64			0.33	3.82%			0.03	0.39%
Overall Cooling Capacity Uncertainty	**%**	**3.7%**								
Overall EER Uncertainty	**%**	**3.8%**								
Outdoor Side (Condenser Coil)										
Barometric Pressure	psia	14.53	-0.1022	0.05	-0.005	-0.07%				
Entering Air Dry Bulb Temp	° F	95.0	0.0159	0.60	0.010	0.12%	0.30	0.0224	0.0004	0.01%
Entering Air Wet Bulb Temp	° F	75.1	-1.5404	0.60	-0.924	-12.08%	0.50	0.0373	-0.0574	-0.86%
Leaving Air Dry Bulb Temp	° F	110.3	0.3918	1.50	0.588	7.68%	0.30	0.0224	0.0088	0.13%
Leaving Air Sample Dry Bulb Temp	° F	105.7	-0.4119	0.30	-0.124	-1.61%	0.40	0.0298	-0.0123	-0.18%
Leaving Air Sample Wet Bulb Temp	° F	80.1	1.7319	0.30	0.520	6.79%	0.40	0.0298	0.0516	0.77%
Nozzle Flow Diff Pressure	in wc	0.738	6.3999	0.02	0.128	1.67%	0.01	0.0009	0.0060	0.09%
Nozzle Flow Static Pressure	in wc	0.031	0.0122	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	110.0	-0.0087	0.50	-0.004	-0.06%				
Nozzle Diameters (3)	in	see note 1	2.1877	0.02	0.044	0.57%				
Nozzle Coefficient	%	see note 2	0.0971	2.00	0.194	2.54%				
Total Power	kW	10.63	-0.2844	0.1063	-0.030	-0.40%				
Air Flow	acfm	4708								
Cooling Capacity (Note 4)	tons	6.69			1.24	18.56%			0.08	1.18%
					Effectivenss Bias Uncertainty					
			(% per change)		(% points)	(% of value)				
Entering Air Dry Bulb Temp	° F	95.0	0.01925	0.60	1.2%	2.18%				
Entering Air Wet Bulb Temp	° F	75.1	0.01769	0.60	1.1%	2.00%				
Evap Pad Outlet Temp	° F	84.5	-0.03694	1.50	-5.5%	-10.44%				
Effectiveness	%	53.1%			5.8%	10.85%				
Overall Cooling Capacity Uncertainty	%	18.7%								
Heat Balance Error	%	12.6%								
Uncertainty in Heat Balance	%	19.1%								

Note 1: The indoor room air flow chamber used two nozzles with following diameters: 7.991, 6.002 inches
The outdoor room air flow chamber used three nozzles with the following throat diameters: 8.998, 8.997, 8.991 inches
Note 2: Nozzle coefficient uncertainty was used to account for possible flow profile effects
Note 3: Cooling Capacity from outdoor side calculations uses total energy change through condenser

Note 4: IDB = indoor dry bulb
IWB = indoor wet bulb
ODB = outdoor dry bulb
OWB = outdoor wet bulb

As seen in the above table, the heat balance uncertainty is significantly larger for this unit. The uncertainty on the indoor side (primary result) has improved, but the uncertainty on the outdoor side is much larger. This is because this unit is adding moisture through the condenser with the evaporative pre-cooler. Because of this, the entering and leaving wet bulb temperature measurements now contribute to the uncertainty in this calculation. Without the pre-cooler, only the dry bulb temperature measurements significantly effect the uncertainty. The heat balance error on this unit varied between about 5% to 15%, and was typically within the calculated uncertainty in the heat balance.

Table VI-5: Summary of Selected Uncertainty Analysis Results for Test Unit #3

Test Designation	A	B	C	Max
Nominal Test Conditions:				
Indoor entering air dry bulb temperature (°F)	80	80	80	80
Indoor entering air wet bulb temperature (°F)	67	67	57	67
Outdoor entering air dry bulb temperature (°F)	95	82	82	115
Cooling Capacity Result (tons)	6.85	6.91	6.56	6.17
Cooling Capacity Uncertainty (±%)	3.8%	3.7%	3.4%	4.1%
Power Result (kW)	10.1	9.02	9.02	12.0
Power Uncertainty (±%)	1%	1%	1%	1%
EER Result (Btu/Wh)	8.15	9.19	8.73	6.19
EER Uncertainty (±%)	3.9%	3.8%	3.5%	4.3%

Below is an example of the uncertainty analysis details for Test Unit #3 for Test Condition “A”.

Table VI-6: Test Unit #3 Uncertainty Analysis for “A” Test Conditions

Nominal Test Condition (IDB/IWB - ODB/OWB) Test # Test Unit #3	see Note 4	80/67 - 95 0812A								
Parameter	Units	Test Value	Sensitivity Coefficient (tons per unit parameter change)	Parameter Bias Uncertainty	Cooling Capacity Bias Uncertainty (tons)	Cooling Capacity Bias Uncertainty (%)	Parameter Standard Deviation	Parameter Precision Index	Cooling Capacity Precision Index (tons)	Cooling Capacity Precision Index (%)
Indoor Side (Evaporator Coil)										
Barometric Pressure	psia	14.59	0.1068	0.05	0.005	0.08%				
Entering Air Dry Bulb Temp	° F	80.0	0.2708	0.30	0.081	1.19%	0.30	0.0224	0.0061	0.09%
Entering Air Dew Point Temp	° F	60.3	0.4948	0.30	0.148	2.17%	0.50	0.0373	0.0184	0.27%
Leaving Air Dry Bulb Temp	° F	61.5	-0.2700	0.40	-0.108	-1.58%	0.30	0.0224	-0.0060	-0.09%
Leaving Air Dew Point Temp	° F	56.3	-0.4338	0.30	-0.130	-1.90%	0.50	0.0373	-0.0162	-0.24%
Nozzle Flow Diff Pressure	in wc	1.908	1.7745	0.02	0.035	0.52%	0.04	0.0030	0.0053	0.08%
Nozzle Flow Static Pressure	in wc	0.399	0.0086	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	62.1	-0.0067	0.50	-0.003	-0.05%				
Nozzle Diameters (2)	in	see note 1	1.9435	0.01	0.019	0.28%				
Nozzle Coefficient	%	see note 2	0.0685	1.00	0.069	1.00%				
Air Flow	acfm	2966								
Cooling Capacity	tons	6.85			0.25	3.68%			0.03	0.39%
EER	Btu/Wh	8.15			0.31	3.81%			0.03	0.39%
Overall Cooling Capacity Uncertainty	**%**	**3.8%**								
Overall EER Uncertainty	**%**	**3.9%**								
Outdoor Side (Condenser Coil)										
Barometric Pressure	psia	14.59	0.2785	0.05	0.014	0.20%				
Entering Air Dry Bulb Temp	° F	95.1	-0.5910	0.60	-0.355	-5.17%	0.30	0.0224	-0.0132	-0.21%
Entering Air Wet Bulb Temp	° F	69.0	-0.0312	0.60	-0.019	-0.27%	0.50	0.0373	-0.0012	-0.02%
Leaving Air Dry Bulb Temp	° F	111.5	0.5288	1.00	0.529	7.71%	0.30	0.0224	0.0118	0.19%
Leaving Air Sample Dry Bulb Temp	° F	102.3	-0.0355	0.30	-0.011	-0.16%	0.50	0.0373	-0.0013	-0.02%
Leaving Air Sample Wet Bulb Temp	° F	70.8	-0.0225	0.30	-0.007	-0.10%	0.50	0.0373	-0.0008	-0.01%
Nozzle Flow Diff Pressure	in wc	1.517	2.9492	0.02	0.059	0.86%	0.02	0.0015	0.0044	0.07%
Nozzle Flow Static Pressure	in wc	-0.182	-0.0198	0.01	0.000	0.00%				
Nozzle Flow Temperature	° F	111.3	-0.0394	0.50	-0.020	-0.29%				
Nozzle Diameters (3)	in	see note 1	2.0345	0.02	0.041	0.59%				
Nozzle Coefficient	%	see note 2	0.0887	2.00	0.177	2.59%				
Total Power	kW	10.09	-0.3157	0.1009	-0.032	-0.46%				
Air Flow	acfm	6740								
Cooling Capacity (Note 3)	tons	6.31			0.67	10.56%			0.02	0.29%
Overall Cooling Capacity Uncertainty	%	10.6%								
Heat Balance Error	%	7.9%								
Uncertainty in Heat Balance	%	11.2%								

Note 1: The indoor room air flow chamber used two nozzles with following diameters: 7.991, 6.002 inches
The outdoor room air flow chamber used three nozzles with the following throat diameters: 8.998, 8.997, 8.991 inches
Note 2: Noxxle coefficinet uncertainty was used to account for possible flow profile effects
Note 3: Cooling Capacity from outdoor side calculations uses sensible energy change through the condenser

Note 4: IDB = indoor dry bulb
IWB = indoor wet bulb
ODB = outdoor dry bulb
OWB = outdoor wet bulb

The heat balance uncertainty improved for Test Unit #3. This was due to better mixing in the condenser outlet temperature measurement location, probably because there were two condenser fans on this unit. The heat balance error on this unit typically varied between 5% to 10%.

The uncertainty estimates provided above are absolute uncertainty estimates. Because many of these uncertainty values are bias's, they would tend to be relatively constant. Therefore, comparisons of performance between test units should have much better accuracy than these absolute uncertainty values.

In addition to providing confidence limits in the results, this uncertainty analysis leads to conclusions regarding improvements which can be made to the measurement system to improve the overall accuracy of the testing. Primarily, better mixing and/or more temperature sensors at the unit discharge locations would improve the leaving temperature measurement accuracy. Also, further investigation into flow profile concerns might lead to improved flow measurement uncertainty.

References

ASME 1998. ASME 19.1-1998. Instruments and Apparatus, Supplement to ASME Performance Test Codes, "Test Uncertainty", 1998.

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